

THERMODYNAMICS

APPLIED TO ENGINEERING

BY THOMAS MACDONALD



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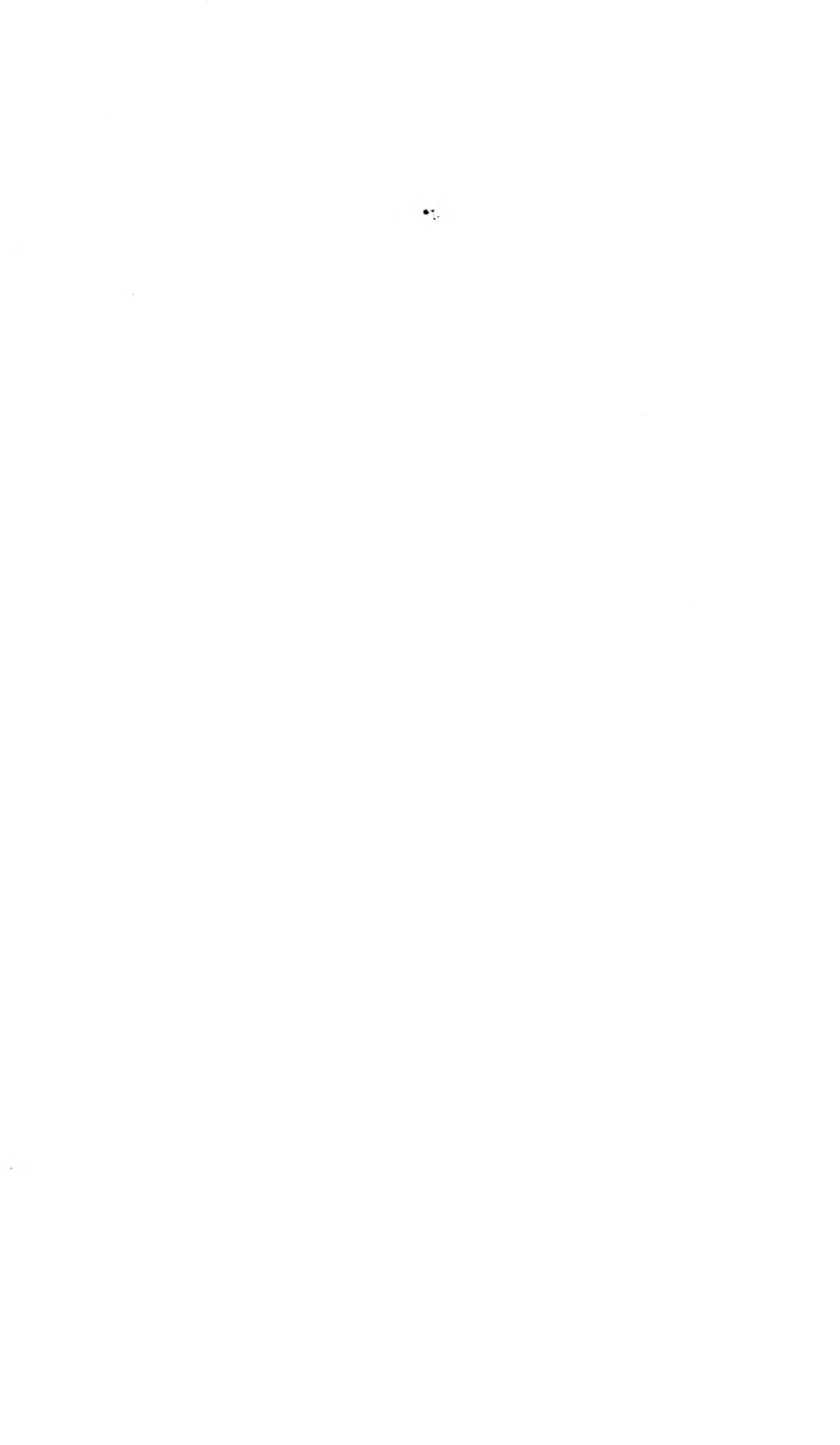
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THERMODYNAMICS
APPLIED TO ENGINEERING

THERMODYNAMICS APPLIED TO ENGINEERING

BY

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PREFACE

THE purpose of this text is to present the principles of engineering thermodynamics in the simplest fashion, and to illuminate these conceptions by reference to the best British and American practice in the major fields of their application. In the selection of examples for illustration stress has been laid on recent developments likely to make a strong imaginative appeal on account of their unusual interest.

The standard of the work is that which is characteristic of University degree requirements on both sides of the Atlantic, and it is hoped that there will be a welcome both in America and Great Britain.

Both as a student and teacher, the author is indebted to Sir J. A. Ewing's lucid exposition of this subject. Thanks are also due to Professor William J. Goudie for suggestions in reference to the determination of the resultant pressure on turbine blading; to the American Society of Mechanical Engineers and the Institution of Mechanical Engineers, London, for permission to use information contained in original papers by members; and to the various firms who have readily supplied information concerning their products. Mr. J. C. MacLagan, designer of the North British double-acting two-cycle Diesel engine, and his firm were particularly helpful in this respect.

THE UNIVERSITY OF VIRGINIA

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SECTION I
GENERAL THERMODYNAMICS

GENERAL THERMODYNAMICS

THE science of thermodynamics, from the point of view of the engineer, is concerned with the transformation of energy in the form of heat into mechanical work, and with the transfer of heat from one body to another.

Machines which give out work as a result of the application of heat to their working substance are termed heat engines, while those which cause a transference of heat from a body, thereby rendering it colder than its surroundings, are called refrigerating machines.

Nature of Heat.—It becomes important, therefore, to inquire into the nature of heat and its relation to matter, before giving consideration to the means whereby the stores of heat energy which exist in coal and oil are turned to useful account.

Heat used to be considered a subtle fluid permeating the interstices of matter. A body was hot or cold according to whether an excess or deficiency of caloric, as this fluid was called, was present. It is now definitely known that heat is molecular kinetic energy in a communicable form.

Application of the Kinetic Theory of Heat to a Gas.—The gaseous working substance of a heat engine consists of a vast number of molecules darting hither and thither at random, the course and velocity of each being determined by encounter with its neighbours.

Between each encounter the free path (as it is termed) may be said to be a straight line, and the molecule to move with uniform velocity. We shall find it convenient to assume that this free path, in the case of a gas, is very large compared with the dimensions of the molecule, and

that the proportion of time taken by encounters is negligible. The energy possessed by a gas, and which may be communicated to other bodies either as a whole (mechanical energy) or as energy of the constituent particles (heat energy), consists of :

- (1) Energy of translation of the molecule ;
- (2) Energy of rotation of the molecule ;
- (3) Energy of vibration of the atoms composing the molecule ; and
- (4) Energy of vibration of the constituent parts of the atom.

(1) and (2) are the most important ; (3) becomes of greater importance as the temperature rises ; while (4) is of small consequence even at high temperatures.

Pressure of a Gas.—Consider any simple gas, such as oxygen. All the molecules will be of the same weight w , let us say.

If there are N molecules per cubic foot, the density of the gas is wN , and the volume per unit mass is $1/wN$. With the simple assumptions already suggested in reference to what we may term an ideal gas, we shall be able to deduce an important relation between the pressure exerted by the gas, its specific volume, and the temperature.

The pressure P , say, in pounds per square foot, which a gas exerts is due to the molecular bombardment of the walls of the containing vessel. In order to determine the magnitude of this pressure we may proceed as follows. Consider any flat area of the surface in contact with the walls, and let the motions of the molecules near it be resolved along three axes at right angles: at any given instant only those molecules having velocity components in the direction of the wall will produce any effect. Such velocity components, on the encounter of the molecule, will be reversed, and the total change of velocity will be twice the magnitude of the original velocity component. We have to remember, in our dealings with the laws relating to gases, that they are statistical statements, and we must agree to the view that if a finite volume, such as a cubic foot, be considered, the number of

molecules which behave in a particular manner will be the same as in any other cubic foot.

Consider, then, the film of gas 1 square foot in cross-section and d of a foot thick, d being very small. In such a film, at any instant, we should have a certain number n of molecules having components of velocity normal to the wall equal to v_{x_1} feet per second, and since the total number of molecules is very large, and motion takes place indifferently in all directions, we shall have an equal number moving in the opposite direction at $-v_{x_1}$ feet per second. Let t be the time taken by a molecule moving at v_{x_1} to traverse the film, without an encounter. Then the number of blows delivered per the time t , by molecules of the group under consideration, will be equal to n , if encounters be neglected, or if the effect of such be negligible. In order to determine the pressure applied by the rapid molecular flow of the selected group through the film, consider the analogous case of a stream of water directed upon a semicircular vane.

The mass = wn , and the acceleration = $2 \cdot v_{x_1}/t$.

Therefore the pressure $P_1 = \frac{wn \cdot 2v_{x_1}}{gt}$.

Again, d being the thickness of the film, and t the time to traverse it,

$$\frac{d}{t} = v_{x_1}.$$

Further, if we think of the flat films that go to make up 1 cubic foot,

$$\frac{2n}{d} = N_1,$$

the total number of molecules in 1 cubic foot having components of $+v_{x_1}$ and $-v_{x_1}$. For in each film there are $2n$ such molecules, and each film being d of a foot thick we obtain the relation already given.

By substitution, P_1 , the pressure due to the group under consideration,

$$= \frac{wN_1d}{2g} \cdot 2v_{x_1} \frac{v_{x_1}}{d} = \frac{wN_1v_{x_1}^2}{g}$$

The total pressure $P = P_1 + P_2 + P_3 + \dots$ etc.

$$= w \{ N_1 v_{x_1}^2 + N_2 v_{x_2}^2 + N_3 v_{x_3}^2 \dots \text{etc.} \} \frac{1}{g} = \frac{wN v_x^2}{g}$$

where N is the total number of molecules per unit of volume, and v_x^2 is the average of $v_{x_1}^2$, $v_{x_2}^2$, etc.

Now consider the relation between the actual velocity of the molecule and its components resolved along three perpendicular axes, x , y , and z . In the xy plane let the projection of the actual velocity v be v_p .

$$\text{Then} \quad v_p^2 = v_x^2 + v_y^2.$$

$$\text{But} \quad v^2 = v_p^2 + v_z^2.$$

$$\therefore \quad v^2 = v_x^2 + v_y^2 + v_z^2.$$

Since the gas as a whole is at rest,

$$v_x^2 = v_y^2 = v_z^2,$$

and therefore $v^2 = 3v_x^2$.

v is not, obviously, the average molecular velocity, but is the velocity which a molecule having average kinetic energy would possess. Hence we arrive at the equation

$$P = \frac{\frac{1}{3}wNv^2}{g}$$

Relation of Pressure, Specific Volume, and Molecular Kinetic Energy.—Since wN is the mass of the gas per cubic foot, the number of cubic feet per pound is given by $\frac{1}{wN}$.

Therefore we have $PV = \frac{\frac{1}{3}v^2}{g}$, where P is in pounds per square foot, and V is in cubic feet per pound.

Temperature.—It now becomes necessary to consider what we mean by the word 'temperature.' Broadly, the term is associated with the idea of flow of heat. We say that if one body be at a higher temperature than another, heat will tend to pass from one to the other, until both are at the same temperature.

We measure the magnitude of this tendency by noting the expansion of mercury, say, referred to such standard conditions as the temperature of melting ice, and the

boiling-point of water, under atmospheric pressure. We may, and do use, for this purpose, hydrogen and other so-called permanent gases, which are far removed from their points of liquefaction.

By maintaining the pressure constant, changes in volume caused by change in temperature may be noted, and some scale adopted by general agreement. The results obtained by the use of such gas thermometers agree very closely amongst themselves, and also, but less exactly, with the scale of the mercury thermometer.

Absolute Temperature.—Taking again the equation

$$PV = \frac{\frac{1}{3}v^2}{g},$$

it becomes apparent that, for the ideal gas we have had under consideration, the temperature, reckoned from the point where V would become zero (were the linear relation of volume and temperature to hold), is proportional to the kinetic energy of translation of the molecule, i.e. $v^2 \propto T$.

We have therefore passed from a measure of temperature which depends upon the expansive properties of some particular substance to a conception of this quantity based upon the amount of energy of translation possessed by the molecules of the ideal gas.

Boyle's Law.—From the above fundamental equations we notice that if the average kinetic energy of translation of the molecule remains constant, the product of pressure and volume remains constant. This is strictly true only of the ideal or perfect gas, and is not exactly true of the permanent gases, while vapours or gases near their point of liquefaction depart widely from it. That this is to be expected may be realized from a consideration of the postulates upon which the equation relating pressure, volume, and molecular kinetic energy is built.

Not only are the dimensions of the molecule supposed to be inconsiderable compared with the free path, but the time occupied by an encounter is supposed to be negligible. If these quantities cannot be neglected, and

encounters occur between molecules whose dimensions are appreciable, the effect of such encounters might be to produce a more rapid bombardment of the walls of the containing vessel; in this case the pressure would be greater, unless this effect were more than counter-balanced by loss of time caused by association during encounters.

Charles' Law.—The experimental fact that actual gases obey the law $V \propto T$ when P is constant, is known as Charles' Law. It follows from our conception of temperature already developed, that the perfect gas obeys this law exactly. The permanent gases, notably hydrogen, follow it very closely, the correction to be applied to a gas thermometer of the constant pressure type, using hydrogen, being of the order of one-sixth of a degree when the temperature has risen to 1800 deg. Fahr., or fallen as low as -250 deg. Fahr.

Avogadro's Law.—In equal volumes of different gases at the same temperature and pressure, we have the same number of molecules.

Considering two gases at the same pressure P ,

$$\frac{\frac{1}{3}w_1N_1v_1^2}{g} = \frac{\frac{1}{3}w_2N_2v_2^2}{g}.$$

Now Maxwell has shown that if two gases be at the same temperature, the average kinetic energy of the molecule in each is the same.

$$\begin{aligned} \text{Hence} \quad \frac{\frac{1}{2}w_1v_1^2}{g} &= \frac{\frac{1}{2}w_2v_2^2}{g} \\ \therefore N_1 &= N_2. \end{aligned}$$

Dalton's Law.—The partial pressure due to the presence of each constituent gas of a mixture is the same as though the other gases were not present.

If the conditions in the mixture are such as we have already imagined in the case of the perfect gas, then the total pressure $P = P_1 + P_2 + P_3 \dots$ etc.

$$\text{i.e.} \quad = \frac{\frac{1}{3}w_1N_1v_1^2}{g} + \frac{\frac{1}{3}w_2N_2v_2^2}{g} \dots \text{etc.}$$

the total pressure being simply the sum of the pressure due to each gas separately.

Joule's Law.—It was shown experimentally by Joule that if a certain quantity of gas contained in one chamber were allowed to expand into another which was empty, without, on the whole, doing any external work, no change in temperature could be detected. Precautions were, of course, taken to prevent the influence of heat transfer from the gas to its surroundings, (or *vice versa*,) affecting the result. This might be deduced from the considerations governing the behaviour of a perfect gas, but could not be assumed in the case of an actual gas, since an appreciable amount of energy might be absorbed in doing work against intermolecular forces during the change in volume.

Experiments by Joule and Thomson, using comparatively large quantities of air and other gases which were caused to expand from one chamber to another by passing them through a porous plug, showed that a small change in temperature actually did take place, and was in general a cooling at ordinary temperatures. Inversion, however, of this effect takes place at high temperatures in most gases, the gas becoming warmer after passing the plug. The temperature of inversion in the case of hydrogen is so low that the normal effect of throttling is a slight warming. As a result of these observations Joule established the law which is known by his name :

‘When a gas expands without doing external work and without taking in or giving out heat, and therefore without changing its stock of internal energy, its temperature does not change.’

Molecular Kinetic Energy of Translation.—An indication has been given that the molecule may possess communicable energy in various forms. We shall now endeavour to form some idea of their relative magnitude in different gases.

Taking again the equation

$$PV = \frac{1}{3} \frac{v^2}{g},$$

we observe that since E_{tr} , the translational energy of a number of molecules totalling unit mass, is equal to $\frac{1}{2} \frac{v^2}{g}$, it therefore follows that $PV = \frac{2}{3} E_{tr}$.

Now we have already seen that $v^2 \propto T$.

Let, then,
$$\frac{\frac{1}{2} v^2}{g} = RT,$$

R being a constant depending upon the properties of the gas and the units employed to measure mass, space, time, and temperature.

Hence $PV = RT$.

This is the fundamental equation relating to gases.

Thus also, $E_{tr} = \frac{3}{2} RT$. When the units are feet, pounds, and seconds, the translational energy is given in foot-pounds per pound.

The Specific Heats of a Gas.—The specific heat of a substance is defined as the amount of heat necessary to raise its temperature one degree under specified conditions, and per unit mass. In the case of a gas there are two important methods of measuring the specific heat, namely, (1) at constant volume, C_v ; (2) at constant pressure, C_p .

In the first case, since the volume does not change, no external work is done and all the heat added goes to increase the stock of internal energy of the gas. In case (2) external work is done, and as a consequence of Joule's law, and, of course, of the principle of the conservation of energy already implicitly assumed, the difference between C_p and C_v gives a measure of this external work.

Ratio of Specific Heats of a Monatomic Gas.—

Suppose that we now consider a monatomic gas whose atoms are incapable of either receiving or communicating kinetic energy of rotation. Then the whole communicable energy possessed by such a gas would consist of energy of translation of the atoms; and thus E_{tr} would equal E_{tot} , the total internal energy.

The change of internal energy, depending as it does

only upon the temperature, for any given temperature change, would be

$$E_{tot_1} - E_{tot_2} = \frac{3}{2}R(T_1 - T_2),$$

consistency of units being carefully observed. For instance, if T_1 and T_2 are in degrees Fahr., and R is the appropriate constant for the gas under consideration, then E_{tot_1} and E_{tot_2} would be expressed in units of mechanical work.

Per degree rise in temperature the change in internal energy is therefore $\frac{3}{2}R$.

This, by definition, is C_v , since, when the volume of a gas remains constant, no external work is done, and all the heat added goes to increase the stock of internal energy.

Now R is also a measure of the external work done when unit mass of a gas is heated under constant pressure through one degree.

$$\begin{aligned} \text{For} \quad & PV_1 = RT_1, \\ \text{and} \quad & PV_2 = RT_2. \\ \therefore \quad & P(V_1 - V_2) = R(T_1 - T_2); \end{aligned}$$

that is, R is a measure of the work done (in foot-pounds, say) per degree rise in temperature.

$$\begin{aligned} \text{Thus} \quad & C_p = C_v + R \\ & = \frac{3}{2}R + R \\ & = \frac{5}{2}R \end{aligned}$$

$$\text{and} \quad \frac{C_p}{C_v} = 1.667.$$

Experiments upon monatomic gases, such as argon and helium, indicate that our assumptions are very approximately applicable to the case of actual gases.

Ratio of Specific Heats of a Diatomic Gas.—As in the case of the monatomic gas, the diatomic molecule has three degrees of freedom of translation; that is, its velocity in any particular direction may be completely defined by reference to three mutually perpendicular axes. We may reasonably infer that in any quantity of gas to which our statistical methods are applicable, the total translational energy may be divided into three equal parts,

each equal to $\frac{1}{2}RT$. Now, in addition to these three degrees of freedom of translation, the molecule may have energy of rotation about any one of three mutually perpendicular axes.

Rotation, however, about an axis through the atomic centres would, let us imagine, have no effect either in transmitting or receiving energy of rotation. Suppose we divide the total kinetic energy equally amongst the five effective degrees of freedom, then

$$E_{tot} = \frac{5}{2}RT$$

and

$$C_p = \frac{5}{2}R + R$$

\therefore

$$\frac{C_p}{C_v} = 1.4$$

This figure is found to agree fairly closely with observed values of γ , the ratio of the specific heats for diatomic gases, such as air, hydrogen, etc., at normal temperatures. While the ratio between the translational and rotational energies remains constant, communicable energy in the form of relative vibration of the constituents of the molecule accounts for the observed increase in the specific heats with increase in temperature.

Suppose, for instance, that at a given temperature E_{tot} had increased from $\frac{5}{2}RT$ its value at normal temperature to $\frac{5}{2}RT_1 + mRT_1$, due to the increase in vibratory motion.

Then γ would be equal to $\frac{\frac{5}{2}R + R + mR}{\frac{5}{2}R + mR} = \frac{7 + 2m}{5 + 2m}$, which shows that while the difference between the specific heats would remain constant, their magnitudes would increase and their ratio be reduced.

Ratio of Specific Heats of a Triatomic Gas.—The values of the specific heats for triatomic and other gases having more than three atoms per molecule may be similarly calculated, but with results which compare less exactly with observation.

This doubtless indicates that even at normal temperatures an appreciable part of the total kinetic energy exists as energy of vibration of the atoms amongst

themselves and as relative movements of the constituents of the atom.

The Transformation of Heat Energy into Mechanical Work.—We shall now consider means of calculating transfers of energy from the molecule to the group, and the nature of the action which takes place when the disordered motion of the molecules is transformed into the ordered motion of, say, the piston and associated linkage of an engine. If the piston of an engine

cylinder containing a gas under pressure be not stationary but move, work is done, for, during every encounter of a molecule with the piston face, the two move together and the molecule recoils with diminished kinetic energy. Thus when a gas is allowed

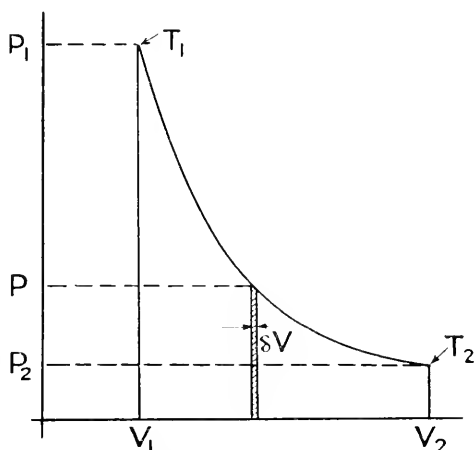


FIG. 1.—Work done during Adiabatic Expansion of a Gas.

to expand without receiving heat from any external source its temperature falls, and this, we know, is an indication of the loss of molecular kinetic energy.

Under such circumstances, and supposing that no energy in the form of heat is lost by the gas, let the pressure of unit mass of a given gas be plotted against its volume as abscissæ.

Then, in order to simplify our ideas, imagine that the containing cylinder has a cross-sectional area of 1 square foot. Clearly, the work done while the piston moves δV foot is $P\delta V$.

Now

$$P\delta V = -\delta E_{tot}$$

i.e.

work done = loss of E_{to}

As already shown, $\delta E_{tot} = C_v \delta T$.

Also,
$$P \delta V = \frac{RT}{V} \delta V.$$

Therefore in the limit,

$$-C_v dT = -\frac{RT dV}{V}.$$

Therefore also
$$-C_v \log_{\epsilon} \frac{T_2}{T_1} = \log_{\epsilon} \frac{V_2}{V_1} (C_p - C_v),$$

since
$$R = C_p - C_v.$$

$$\therefore -\log_{\epsilon} \frac{T_2}{T_1} = \left(\frac{C_p}{C_v} - 1 \right) \log_{\epsilon} \frac{V_2}{V_1},$$

or
$$\gamma \log_{\epsilon} \frac{V_2}{V_1} - \log_{\epsilon} \frac{V_2}{V_1} = -\log_{\epsilon} \frac{T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad (a)$$

Now
$$P_1 V_1 = RT_1$$

and
$$P_2 V_2 = RT_2$$

$$\therefore \frac{P_2 V_2}{P_1 V_1} = \frac{T_2}{T_1}$$

and
$$\log_{\epsilon} \frac{P_2}{P_1} + \log_{\epsilon} \frac{V_2}{V_1} = \log_{\epsilon} \frac{T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad (b)$$

Adding (a) and (b) together,

$$\log_{\epsilon} \frac{P_2}{P_1} + \gamma \log_{\epsilon} \frac{V_2}{V_1} = 0.$$

$$\therefore \frac{P_2}{P_1} = \left(\frac{V_1}{V_2} \right)^{\gamma}$$

$$\therefore P_1 V_1^{\gamma} = P_2 V_2^{\gamma} = P_3 V_3^{\gamma} \quad . \quad . \quad . \quad \text{etc.}$$

This indicates that under the conditions which we have supposed, the gas follows a law of the form PV^n equals a constant, n having a numerical value equal to the ratio of the specific heats. We may, for the sake of simplifying what follows, make the assumption that the molecules of the perfect gas are capable of possessing energy of rotation and translation only. If (as already suggested) there is an equipartition of energy amongst the various

degrees of freedom of which the molecule is capable, then C_v is constant, and so also is C_p , since R is constant for a perfect gas, and the ratio of the specific heats is constant and independent of the temperature.

With this proviso we may apply the equation relating the pressure and volume of a perfect gas which is expanding and doing work at the expense of its internal energy, to the calculation of the total amount of heat turned into mechanical work between any assigned limits.

$$\begin{aligned}
 \text{Work done} &= \int_{V_1}^{V_2} P dV \quad \text{and} \quad P = \frac{P_1 V_1^\gamma}{V^\gamma} \\
 \therefore \quad \text{,,} \quad \text{,,} &= \frac{P_1 V_1^\gamma \{V_2^{1-\gamma} - V_1^{1-\gamma}\}}{1-\gamma} \\
 &= \frac{P_2 V_2^\gamma V_2^{1-\gamma} - P_1 V_1^\gamma V_1^{1-\gamma}}{1-\gamma} \\
 &= \frac{P_1 V_1 - P_2 V_2}{\gamma - 1}
 \end{aligned}$$

Since we are dealing with unit mass of the gas, this equation gives the work done in foot-pounds per pound.

For any other mass of gas the appropriate values of V_1 and V_2 would be substituted.

Adiabatic Expansion.—Such a mode of expansion (or compression) as above outlined, in which all the work done by, or upon, the gas decreases (or increases) its total stock of internal energy, none being allowed to pass through the walls of the containing vessel in the form of heat or to return to the gas through eddy motion, is known as an adiabatic process.

A gas may, of course, expand in an infinite number of ways, depending upon the extent of the interchange of heat. We have just dealt with that case in which there is no interchange, the gas expanding and doing work at the expense of its internal energy. In the other important mode the temperature does not change, all the work being done at the expense of the heat added during expansion. This is known as an

Isothermal Expansion.—Since the gas will, therefore, obey Boyle's law, PV equals a constant and the work done is as before—

$$\int_{V_1}^{V_2} PdV = P_1 V_1 \log_e \frac{V_2}{V_1}.$$

This may also be written in the form $RT_1 \log_e \frac{V_2}{V_1}$.

Before proceeding to consider how the work done by an expanding gas may be turned to useful account in some form of engine, it will be necessary to make two important observations, one of which is a particular case of the principle of the conservation of energy ; while the other denies the possibility of causing heat to pass from one body to another at a higher temperature without the intervention of some form of power-driven machine. These are known as the laws of thermodynamics.

The First Law of Thermodynamics.—When mechanical energy is produced from heat a definite quantity of heat goes out of existence for every unit of work done, and *vice versa*.

The Second Law of Thermodynamics.—It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.

No exception to these laws has, so far, been discovered, and they rank with the other primary concepts concerning the universe which have been established by experience. As far as the First Law is concerned, there is no reason why the total amount of heat possessed by a body should not be turned into mechanical work. The Second Law, however, sets a definite limit to the efficiency of a heat engine working under assumed conditions, and enables us to set up a standard by which the performance of actual heat engines may be compared.

Action of a Heat Engine.—Essentially in all heat engines the working substance, which may be a gas, a mixture of a vapour and its liquid, or even a solid body (such as a bar of steel or copper), passes through a cycle

of operations, which consists in the taking in of heat, the transformation of a proportion into mechanical work, and the rejection of the remainder. At the end of the cyclic process the working substance is in exactly the same condition as at the commencement.

Our immediate objective is to discover in what manner we can obtain a maximum amount of work for a given expenditure of heat under assigned conditions.

Conditions of Maximum Efficiency.—Suppose we have two engines A and B (fig. 2), one of which, A, is reversible, that is to say, will work as a heat pump when supplied with power by B.

In order to free the argument from all extraneous issues, let there be no heat or other losses in the process. Then in one cyclic action A takes Q units of heat, transforms a part W into mechanical work and rejects $Q-W$ units to the receiver.

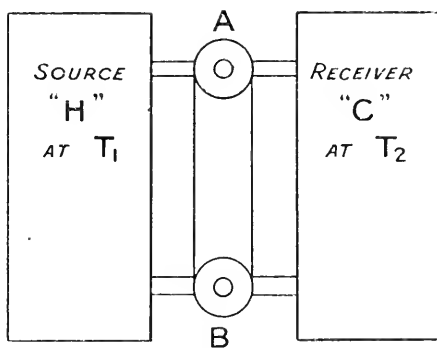


FIG. 2.

When reversed, $Q-W$ units of heat are taken from the receiver, and Q units are delivered to the source H. Both source and receiver are supposed to have large capacities for heat such that the withdrawal or addition of quantities of the magnitude considered has no effect upon the temperatures T_1 and T_2 .

In driving A, since friction is neglected, B delivers W units of heat in the form of mechanical work. Now, if it were possible for B to be more efficient than A, the net result would obviously be that the source H would, on the whole, gain heat at the expense of C, the cool body. This, however, would constitute a violation of the Second Law, and hence we are forced to the conclusion that no heat engine can have a greater efficiency than a reversible

engine when both work between the same limits of temperature.

As will be shown immediately, the measure of this ideal efficiency is given by $\frac{T_1 - T_2}{T_1}$, in which T_1 is the absolute temperature of the working substance during the reception of heat, and T_2 is the absolute temperature of the working substance during the rejection of heat.

Carnot's Cycle.—Just over a hundred years ago Sadi Carnot pointed out that a reversible engine had the maximum attainable efficiency, and explained how such an engine might be made to work if certain simple conditions were assumed. Carnot, however, was unaware of the principle of the conservation of energy, and hence was unable to tell what the efficiency was. In order to calculate it we may imagine an engine which follows Carnot's cycle and which is reversible in the sense in which he understood it.

Let unit mass of a perfect gas be enclosed in a perfectly non-conducting cylinder fitted with a frictionless non-conducting piston. Let there be also supplied a warm body having such a large capacity for heat in comparison with the working substance during expansion that an infinitesimal drop in temperature only takes place: a cold body similarly of large capacity, and a non-conducting cylinder cover, any one of which may be applied to the cylinder end, supposedly of conducting material. The action then proceeds as follows:

In the first stage the gas expands isothermally, in contact with the source of heat at T_1 until the point (ii) is reached (fig. 3).

Since the temperature of the gas, which is perfect, remains constant, the internal energy also is constant. Hence all the work which is done, is done at the expense of the heat added, viz.:

$$\int_{V_1}^{V_2} P dV = RT_1 \log_e r.$$

In the second stage the gas expands adiabatically from

T_1 to T_2 , the non-conducting cover being in place on the cylinder end.

Stage 3 is similar to stage 1, the gas rejecting heat, however, to the receiver, instead of receiving it. Stage 4 is an adiabatic compression which restores the gas to its original condition. It may be noted that the isothermal compression of stage 3 must be stopped at the appropriate point in order that the adiabatic through (iv) may pass through (i).

Now such a cycle is reversible, since it is only necessary

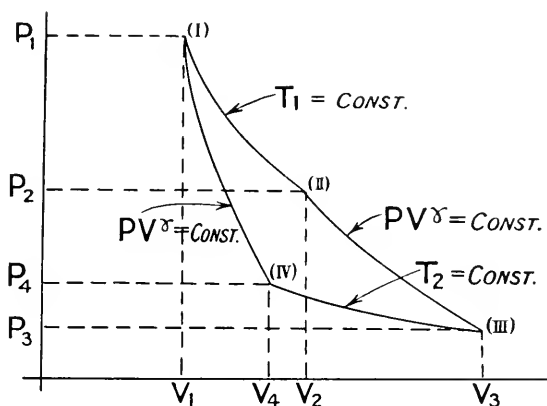


FIG. 3.—Carnot's Cycle.

to think of the temperature during the isothermal expansion from (iv) to (iii) as being less by an infinitesimal quantity than the cold body (instead of greater), so that heat will flow in the opposite direction, and slightly greater (instead of less) during the isothermal compression from (ii) to (i) than the warm body, so that the heat may be delivered. The adiabatics, of course, are reversible for a perfect gas, which is supposed to be expanding so slowly that no losses occur through friction or eddy motion.

The total work done during the four stages is given by

$$P_1 V_1 \log_e \frac{V_2}{V_1} + \frac{P_2 V_2 - P_3 V_3}{\gamma - 1} + P_3 V_3 \log_e \frac{V_4}{V_3} + \frac{P_4 V_4 - P_1 V_1}{\gamma - 1}$$

c

Now $P_2V_2^\gamma = P_3V_3^\gamma$ and $P_1V_1^\gamma = P_4V_4^\gamma$;

also $P_1V_1 = P_2V_2$. $\therefore \frac{P_2}{P_1} = \frac{V_1}{V_2}$;

and similarly $P_3V_3 = P_4V_4$. $\therefore \frac{P_3}{P_4} = \frac{V_4}{V_3}$.

Hence $\left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{V_3}{V_4}\right)^{\gamma-1}$ $\therefore \frac{V_2}{V_1} = \frac{V_3}{V_4} = r$ say.

Adding up these four terms we have, for the net work done, $P_1V_1 \log_e r - P_2V_2 \log_e r$, and thus the efficiency becomes

$$\begin{aligned} \frac{P_1V_1 \log_e r - P_2V_2 \log_e r}{P_1V_1 \log_e r} &= \frac{RT_1 \log_e r - RT_2 \log_e r}{RT_1 \log_e r} \\ &= \frac{T_1 - T_2}{T_1}. \end{aligned}$$

Lord Kelvin's Scale of Absolute Temperature.—

Imagine a series of Carnot engines, each of which gives out W units of work. Let number one take in Q units from the source at T_1 , and reject the balance of the heat to engine number two at T_2 ; these temperatures being defined by reference to the perfect gas used as the working substance. Let now number two take in heat at T_2 and reject at T_3 , and so on.

Then number one does $\frac{Q(T_1 - T_2)}{T_1}$ units of work (if, of course, the heat taken in is expressed in work units) and rejects $\frac{QT_2}{T_1}$ units. Number two takes in $\frac{QT_2}{T_1}$ units and does $\frac{QT_2}{T_1} \cdot \frac{(T_2 - T_3)}{T_2}$ units of work. Since, by arrangement,

$$\frac{Q(T_1 - T_2)}{T_1} = \frac{Q(T_2 - T_3)}{T_1}$$

$\therefore T_1 - T_2 = T_2 - T_3 = T_3 - T_4 \dots$ etc.

We have here, therefore, a new conception of temperature which does not depend upon the properties of any substance whatsoever, but which is based upon the unit of work.

Further, the intervals of temperature so defined by reference to a certain quantity of work are equal to the corresponding intervals upon the scale of the perfect gas thermometer. This is known as Lord Kelvin's scale.

Entropy.—The consideration of the Carnot cycle, and the resulting information we have gathered respecting the availability of the heat content of a substance for conversion into work, lead us to the idea of entropy. This is a function of the state of the substance, changes being measured by the ratio of quantity of heat entering or leaving the body during a reversible process to the absolute temperature at which such transfer took place. The substance may not only change its entropy, due to the addition or abstraction of heat from the outside, but any irreversible action, causing a transformation from ordered to disordered motion of the constituents of the body due to friction and eddy motion, causes a change.

Thus, for example, in the Carnot cycle, if Q units of heat are added at T_1 , then $\frac{Q}{T_1}$ gives a measure of the change of entropy suffered by the working substance during the isothermal expansion. It is to be particularly observed that while the internal energy of the perfect gas which constitutes the working substance remains constant in the first stage, the departure of a quantity of mechanical work exactly equivalent to the heat taken in produces no entropy change.

If $Q \frac{T_2}{T_1}$ units of heat are rejected in the third stage, then the change of entropy is $Q \frac{T_2}{T_1} \frac{1}{T_2} = \frac{Q}{T_1}$ as above.

During the two adiabatic processes, which are supposedly reversible, no change of entropy takes place, since no heat is allowed either to enter or leave the gas. We observe, therefore, that in the reversible cyclic process of Carnot the total change of entropy is zero, if the appropriate signs be considered. This is true of all reversible cycles. For in any cycle which is not reversible the substance will absorb heat due to internal actions,

of which no account has been taken in the estimation of entropy changes. Hence the summation will give a negative result due to the rejection of heat formerly appearing as ordered motion.

Temperature-Entropy Diagrams.—It is often found convenient to represent the relationship between the temperature and entropy of a substance in the form of a diagram drawn between rectangular co-ordinates.

With such an arrangement the Carnot cycle would take the form of a simple rectangle.

If the temperature of the substance varied during the

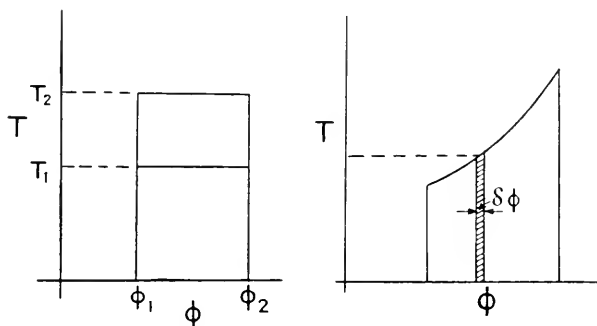


FIG. 4.—Temperature-Entropy Diagrams.

transfer of heat, then in order to determine the entropy change we should have a summation of the form

$$\int \frac{dQ}{T} = \sigma \int_{T_1}^{T_2} \frac{dT}{T}$$

where σ = specific heat of the substance, and the resulting curve would be logarithmic.

Since $\frac{dQ}{T} T = dQ$, therefore the area of the thin vertical strip of the $T\phi$ figure is a measure of the heat added from outside, in those cases where no irreversible action has taken place.

The sum of these quantities, represented by the whole area under the curve, gives the total heat so added. In any cyclic process, therefore, involving only reversible actions, the enclosed area, representing as it does the

difference between the heat added and the heat rejected, indicates, to some scale, the work done.

Actual Engine Cycles having a Gas as the Working Substance.—We are now in a position to consider a few actual heat-engine cycles using a gas, or rather a mixture of gases, as the working substance. In such cases the heat is not, in general, added from the outside, but is generated within the substance itself by combustion of certain of its constituents.

This makes no essential difference to the discussion, and while chemical changes take place, ~~which~~ rendering the process not strictly cyclic, the ordinary gas or Diesel engine is, in the main, a nitrogen motor.

Stirling's Regenerative Cycle.

One of the earliest attempts to reproduce a reversible cycle practically is due to Robert Stirling (1827). His design included a device known as a regenerator, which in more recent applications of this

principle consists of a nest of brickwork, having a temperature gradient along the path of the gases. For our present purpose it may suffice to think of it as being composed of a group of tubes maintained at T_1 at one end, and T_2 at the other, with a continuous fall in temperature between.

The action permitted an isothermal expansion of the gas at T_1 (the temperature of the furnace), followed next by a passage through the regenerator, from T_1 to T_2 , when

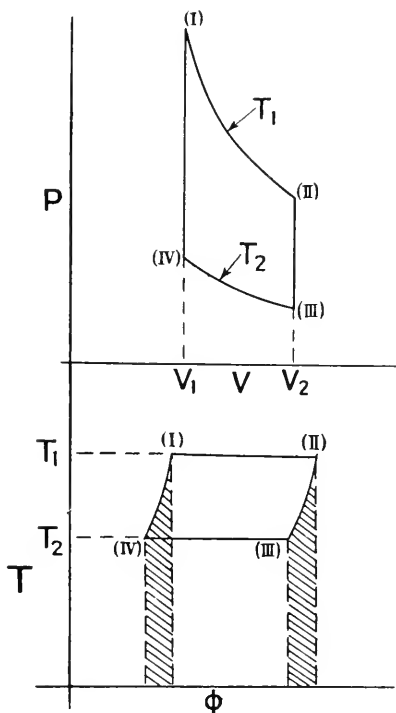


FIG. 5.—Stirling's Regenerative Cycle.

the working substance parted with heat to the walls of the tubes at constant volume. Ideally, this latter process would be reversible, since the temperature of the gas might be supposed to differ by an infinitesimal quantity from that of the tubes at any point. The third stage was an isothermal compression of the same ratio as the expansion, and the fourth a heating at constant volume in the regenerator from T_2 to T_1 .

The pressure-volume diagram is as shown, and the efficiency is given by

$$\frac{RT_1 \log_e r - RT_2 \log_e r}{RT_1 \log_e r} = \frac{T_1 - T_2}{T_1},$$

since the heat taken from the regenerator is equal to that restored.

The temperature-entropy diagram would have the form indicated. It is interesting to observe that though all of the heat received is not added at the maximum temperature, that portion which is used is so added, all the heat absorbed at rising temperature being returned to the regenerator.

Constant Pressure Cycle.—For practical reasons, gas-engines in the main do not follow cycles which are even approximately reversible in the Carnot sense. We may usefully make comparison of a few in common use, noting their divergence from the foregoing. For the sake of simplicity, let the working substance in each case be air, attributing to it the characteristics of a perfect gas.

Consider, then, that case in which the cycle is bounded by two adiabatics and two lines of constant pressure.

Fuel, it may be imagined, is admitted at such a rate as to keep the pressure constant from (i) to (ii). Hence the temperature will rise during combustion, and all of the heat will be added less advantageously than if it were communicated wholly at the highest temperature reached. It is important to realise that actually (as well as ideally) the expansion of the gas is due to the addition of heat and consequent rise in temperature. The gas during this stage obeys Charles' law. After cut-off of the fuel supply the gas expands adiabatically to

atmospheric pressure, let us say. This might be carried out in the cylinder of an engine, but (since expansion is complete) more readily in the nozzle of the gas-turbine.

The stage (iii) (iv) is a cooling at constant pressure which, in the actual engine, takes place in the atmosphere.

A fresh charge of air is then compressed adiabatically from (iv) to (i), and this completes the cycle.

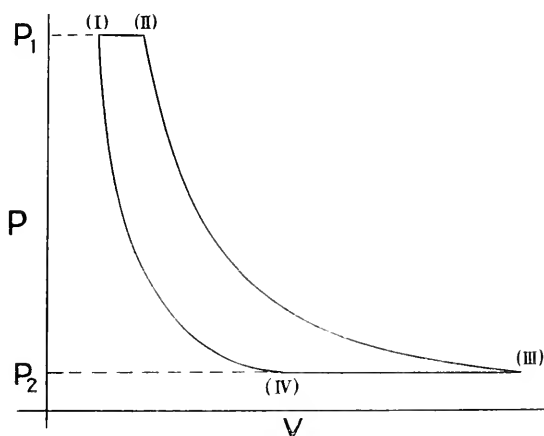


FIG. 6.—Constant Pressure Cycle.

The ideal efficiency is given by

$$\frac{C_p (T_2 - T_1) - C_p (T_3 - T_4)}{C_p (T_2 - T_1)} = 1 - \frac{T_3 - T_4}{T_2 - T_1}$$

Now
$$\frac{T_3}{T_2} = \left(\frac{V_2}{V_3} \right)^{\gamma-1}.$$

Also
$$P_1 V_1^\gamma = P_4 V_4^\gamma \text{ and } P_2 V_2^\gamma = P_3 V_3^\gamma,$$

since
$$P_1 = P_2 \text{ and } P_3 = P_4,$$

$$\therefore \frac{V_1}{V_2} = \frac{V_4}{V_3} \text{ or } \frac{V_2}{V_3} = \frac{V_1}{V_4}.$$

Let
$$\frac{V_3}{V_2} = \frac{V_4}{V_1} \text{ be called } r.$$

Then
$$\frac{T_3}{T_2} = \frac{T_4}{T_1} = \left(\frac{1}{r}\right)^{\gamma-1}.$$

Hence efficiency
$$= 1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

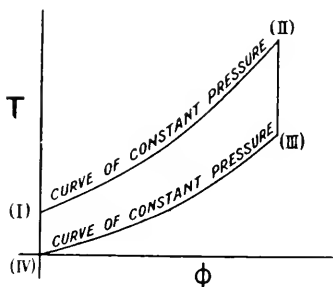


FIG. 7.—Temperature - Entropy Diagram for Constant Pressure Cycle.

The temperature-entropy diagram has the shape sketched.

Constant Volume Cycle.—This is the ideal standard for the ordinary petrol and other engines of similar type. The PV diagram is shown in fig. 8.

Compression of the charge occurs between (iii) and (iv), followed by ignition and the addition of heat at

constant volume. As in the constant pressure cycle, the temperature rises continuously and there is a progressive

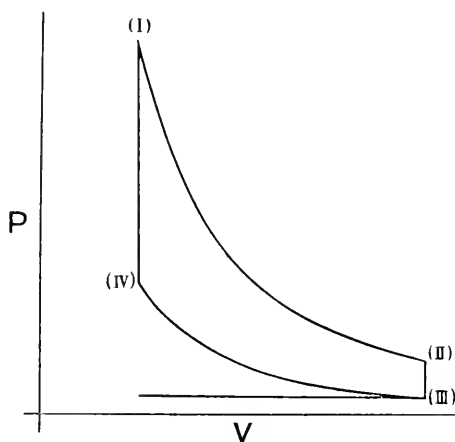


FIG. 8.—Constant Volume Cycle.

increase in the availability of the heat energy so added. If we assume the specific heat at constant volume to be constant, the efficiency is given by

$$\frac{C_v (T_1 - T_4) - C_v (T_2 - T_3)}{C_v (T_1 - T_4)} = 1 - \frac{T_2 - T_3}{T_1 - T_4}.$$

Now $\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1}$ and since $V_1 = V_4$ and $V_2 = V_3$

$$\therefore \frac{T_3}{T_4} = \frac{T_2}{T_1} = \left(\frac{1}{r}\right)^{\gamma-1} \text{ if } r = \frac{V_3}{V_4} = \frac{V_2}{V_1}.$$

Hence efficiency $= 1 - \left(\frac{1}{r}\right)^{\gamma-1}$ as in the constant pressure cycle.

The temperature-entropy diagram also is similar in form to that of the constant pressure cycle, the constant volume curves, however, having a greater slope.

Diesel Engine Cycle.—This is a modification of the constant pressure type. It has the advantage over the constant volume cycle of rendering practical a much larger ratio of compression; hence both the temperature at the commencement of combustion on the admission of the fuel and the maximum temperature reached are higher. The thermodynamic efficiency is therefore greater.

The following analysis, on the air standard, may serve to indicate the influence of the ratios of compression and expansion, during combustion, upon the efficiency.

$$\text{The heat taken in} = C_p (T_2 - T_1).$$

$$\text{The heat rejected} = C_v (T_3 - T_4).$$

$$\begin{aligned} \text{The efficiency} &= 1 - \frac{C_v (T_3 - T_4)}{C_p (T_2 - T_1)} \\ &= 1 - \frac{(T_3 - T_4)}{\gamma (T_2 - T_1)}. \end{aligned}$$

Now $\frac{T_3}{T_2} = \left(\frac{V_2}{V_3}\right)^{\gamma-1}$ and $\frac{T_4}{T_1} = \left(\frac{V_1}{V_4}\right)^{\gamma-1}$

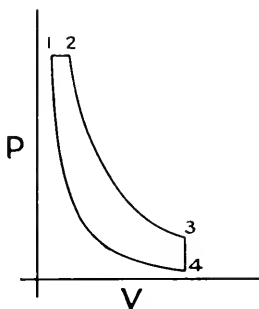


FIG. 9.—Diesel Engine Cycle.

Hence the efficiency becomes

$$\eta = \frac{T_2 \left(\frac{V_2}{V_3}\right)^{\gamma-1} - T_1 \left(\frac{V_1}{V_4}\right)^{\gamma-1}}{\gamma (T_2 - T_1)}$$

Now, since the gas obeys Charles' law in the stage (i) to (ii)

$\frac{V_2}{V_1} = \frac{T_2}{T_1}$, and thus the efficiency

$$= \eta = \frac{\frac{V_2}{V_1} T_1 \left(\frac{V_2}{V_3}\right)^{\gamma-1} - T_1 \left(\frac{V_1}{V_4}\right)^{\gamma-1}}{\gamma \left(T_1 \frac{V_2}{V_1} - T_1\right)}$$

$$\text{giving } \eta = \frac{\frac{V_2}{V_1} \left(\frac{V_2}{V_3}\right)^{\gamma-1} - \left(\frac{V_1}{V_4}\right)^{\gamma-1}}{\gamma \left(\frac{V_2}{V_1} - 1\right)}$$

Call $V_3 = V_4$, V_c ; then, dividing numerator and denominator of the second term by $\left(\frac{V_1}{V_c}\right)^{\gamma-1}$ we have

$$\eta = \frac{\frac{V_2}{V_1} \left(\frac{V_2}{V_c}\right)^{\gamma-1} \left(\frac{V_c}{V_1}\right)^{\gamma-1} - 1}{\gamma \left(\frac{V_2}{V_1} - 1\right) \left(\frac{V_c}{V_1}\right)^{\gamma-1}}$$

$$= \eta = \frac{\left(\frac{V_2}{V_1}\right)^{\gamma} - 1}{\left(\frac{V_c}{V_1}\right)^{\gamma-1} \gamma \left(\frac{V_2}{V_1} - 1\right)}$$

Call $\frac{V_c}{V_1} = r$ and $\frac{V_2}{V_1} = r_1$

then efficiency $= \eta = \left(\frac{1}{r}\right)^{\gamma-1} \frac{r_1^{\gamma} - 1}{\gamma (r_1 - 1)}$.

This indicates that if r_1 be maintained constant, the efficiency increases with increase in the ratio of compression.

Reversal of the Carnot Cycle.—If a Carnot engine were reversed, it would function as a cooling machine. For, consider the effect of placing the cool body in contact with the cylinder during the isothermal expansion from (iv) to (iii). Heat amounting to $RT_2 \log_e r$ would be abstracted and a quantity equal to $RT_1 \log_e r$ delivered to the warm body. The ratio of heat abstracted from the cold body to that spent in work would then be

$$\frac{RT_2 \log_e r}{RT_2 \log_e r - RT_1 \log_e r} \text{ which gives } \frac{T_2}{T_1 - T_2}.$$

Similar reasoning to that already used to prove that no engine can be more efficient than a reversible engine,

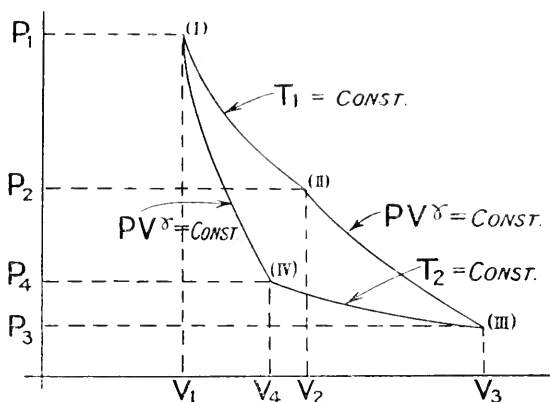


FIG. 10.—Reversal of Carnot's Cycle.

when both work between the same limits of temperature, indicates that no refrigerating machine can have a higher coefficient of performance (as the above ratio is termed) than that of the reversed Carnot cycle.

Cycle of Refrigerating Machine using Air.—In the Bell-Coleman type of machine air is drawn from the chamber to be kept cool, compressed adiabatically to such a temperature that it may readily part with heat to the circulating water, and thereafter expanded to the original pressure, the temperature falling much below that of the air in the cold chamber.

The pressure-volume and temperature-entropy diagrams are shown below.

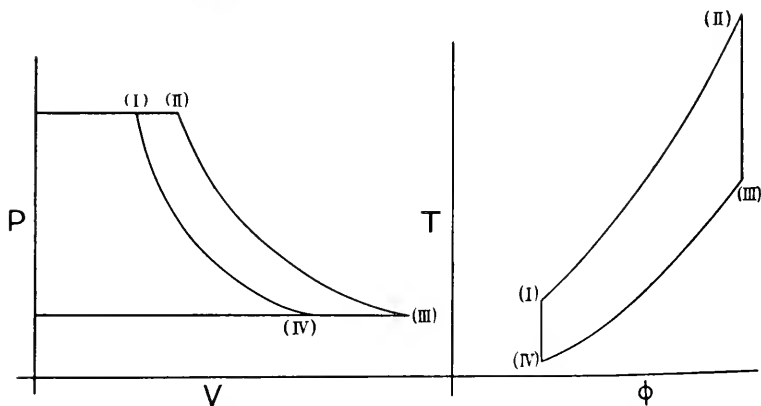


FIG. 11.—Cycle of Refrigerating Machine using air.

Vapours and Liquids.—In order to have a physical basis for our discussion of general thermodynamical principles, and their applications to a few characteristic cycles of engines and refrigerating machines using a gaseous working substance, we imagined the perfect gas, differing but little in its simple characteristics from those of actual gases. It now becomes necessary to study the properties of liquids and their vapours, in order to extend the scope of our investigations.

When a gas is cooled it passes from that state in which its characteristics closely resemble those of a perfect gas to the condition of a vapour, and if the abstraction of heat be continued will ultimately condense and may finally solidify. Thus hydrogen condenses at -402 deg. Fahr., and solidifies at -434 deg. Fahr. Nitrogen condenses at -231 deg. Fahr., and helium, one of the most refractory gases, within a few degrees of absolute zero. The premises adopted in order to deal simply with gaseous working substances will not answer in the case of engines using vapours, which depart too widely in their properties from those of a perfect gas.

It may be remembered that the assumptions made included a supposition that the time occupied by

encounters was negligible ; also that during the free path the molecule moved uniformly in a straight line. If the temperature of a gas, which may be described as a superheated vapour, be lowered sufficiently, intermolecular forces become of practical consequence, coaggregation—that is, a temporary association of molecules forming a group—takes place, and the influence of encounters in altering the total pressure cannot be neglected. The gas is now said to be a vapour, and further lowering of the temperature causes condensation to commence. In the liquid state, while the molecule still has freedom of translation, the period of the free path is negligible in comparison with the time during which the molecule is under the action of intermolecular forces. From the behaviour of liquids in general and water in particular, it would appear that great resistance is offered to the closer approach of the molecules to each other, while a certain, much smaller resistance is experienced when an effort is made to separate them. To such intermolecular forces are ascribed the effects of surface tension, capillarity, the spherical form of drops, etc. ; the phenomena of supersaturation are also ascribed thereto. Since water is of the greatest practical consequence as the working substance of steam engines and turbines, we may profitably study its properties, regarding it as typical of other liquids.

Evaporation.—From all exposed water surfaces, and even from ice, evaporation takes place. It appears that at all temperatures below boiling-point there is an interchange of molecules between the liquid and its surroundings, whenever possible. When the number of molecules which escape from the liquid just balances the number which return the net effect is zero, and the space in the neighbourhood of the water is said to be saturated with water vapour. Dalton's law, which was observed to hold in the case of a mixture of gases, is very approximately true also in the case of a mixture of vapours and gases, except at very high pressures.

The addition, therefore, of a given quantity of water vapour to the space above an exposed water surface increases the pressure upon it by the weight of the vapour

column. If air also is present, then the total pressure is the sum of the pressure of the air and that of the vapour. If the temperature of the air and water vapour mixture rises, evaporation will continue until equilibrium is again restored. The relationship which subsists between vapour pressure and temperature is known, and may be found by reference to tables of the properties of steam.

Thus, at freezing-point, the pressure of the saturated vapour upon an open water surface is 0.0892 pound per square inch. As the temperature rises the vapour pressure rises rapidly, until at 212 deg. Fahr. it is just equal to that of the atmosphere under normal conditions. If the temperature be lowered, the reverse effect is observed. There is now a greater number of molecules returning to the liquid state, and dew forms upon any exposed surfaces, or collects as a fine suspended mist, upon dust particles, or if these be not present in sufficient quantity, upon coaggregated molecules of the vapour itself.

Boiling.—When the pressure which steam would exert at the saturation point just exceeds that of the restraints upon the liquid, evaporation commences at a number of centres within the liquid itself, generally about tiny bubbles of occluded air, and the water is said to boil. The cooling effect of this evaporation throughout the mass of the liquid is, in general, sufficiently effective to prevent further appreciable rise of temperature. (This, however, may happen if suitable nuclei are absent, due to molecular cohesion preventing the formation of bubbles of very small diameter.) When, therefore, the total pressure upon the water is known, the temperature at which boiling will commence may be determined by reference to the tables above mentioned.

Absorption of Heat during the Formation of Steam under Constant Pressure.—If the temperature of unit mass of water be raised, and the amount of heat added during the process be noted, it will be found that while the specific heat is nearly constant there is a steady though small increase after the minimum value, at 86 deg.

Fahr., is passed. It has been suggested that this increase may be due to the absorption by the liquid of saturated vapour.

For most practical purposes, the specific heat of water at moderate temperatures may be taken as constant and equal to the average value between 32 and 212 deg. Fahr. At some point determined by the pressure, boiling commences and thereafter the addition of heat is unaccompanied by any further rise in temperature: in the phraseology of the earlier writers it is said to become latent. When the last particle of moisture has become steam, and before the temperature has changed, the steam is said to be dry and saturated. Any further addition of heat causes the dry steam to become superheated, the temperature again rising.

To find the relationship between any assigned condition of the steam and its total heat (that is, the amount of heat added, together with any work which may have been done, in order to bring it to that condition under constant pressure from any arbitrarily selected state), steam tables may again be consulted. The initial condition usually chosen is that of water at 32 deg. Fahr. In order to visualize the circumstances under which these changes of state may be supposed to take place, it may be helpful to think of the unit mass of water as being introduced under a piston loaded so as to produce the desired constant pressure. To the quantity of heat actually transferred must be added a small quantity which takes account of the work done in introducing the water below the piston.

Thus, if H is the heat added during the formation of unit mass of dry steam and $\frac{PV^w}{J}$ is the heat equivalent of the work done in introducing the water below the piston, the total heat (as it is termed) is the sum of these two quantities.

If the total heat so defined be denoted by I , then

$$I = \frac{PV^w}{J} + H.$$

H, of course, includes the heat added during the rise in temperature of the water (H_w) and that absorbed during the formation of the dry steam (L). That portion of the total energy transferred to the steam which was spent in doing external work is obviously $\frac{PV_s}{J}$, where V_s is the volume of unit mass of dry and saturated steam at pressure P. The balance goes to increase the internal energy E.

Thus we have

$$I^s = E + \frac{P}{J} (V_s - V_w) + \frac{PV_w}{J} = E + \frac{PV_s}{J}.$$

If the steam be wet we have, instead of

$$\begin{aligned} I_s &= I_w + L, \text{ for dry steam,} \\ I_{sw} &= I_w + qL, \end{aligned}$$

where q represents the proportion of vapour in the mixture of moisture and vapour.

If the steam be superheated, then

$$I' = I_w + L + k(t' - t),$$

where k is the specific heat of superheated steam at constant pressure (a quantity which varies both with the pressure and with the degree of superheat), t' is the temperature of superheat, and t is the temperature of formation of the steam. Total heat is generally reckoned from that of water at 32 deg. Fahr., and under the assigned constant pressure.

Entropy of Water and Steam.—During the heating of the water from any absolute temperature T_0 to T_1 the

change in entropy is measured by $\int_{T_0}^{T_1} \frac{dT}{T}$ if the specific

heat of water be taken as constant and equal to unity. During the formation of unit mass of dry steam the change is $\frac{L_1}{T_1}$ where L_1 is the latent heat at the temperature of formation T_1 .

If the steam be superheated, then the change which occurs between any temperature T_1 and temperature of superheat T' is $k \int_{T_1}^{T'} \frac{dT}{T}$, k being the average value of

the specific heat of superheated steam for the appropriate range. A chart which exhibits the relation between temperature and entropy for water and steam is of service in the determination of the condition of the steam after a change of state under assigned conditions.

Isothermal Expansion of Steam.—Water is nearly incompressible. A slight change in volume with change in pressure does, however, take place, this change becoming greater the higher the temperature at which the change in pressure takes place. If, then, we consider the relationship between the pressure and volume of water at any assigned temperature, we should have a curve which at first is nearly vertical, as shown in fig. 12. If the pressure be allowed to fall to that value at which steam will commence to form, expansion will take place under constant pressure so long as we continue to add heat. When all the moisture present has been converted into steam, the steam is dry and further expansion at constant temperature will cause the curve to enter the region of superheat on the pressure-volume diagram. This arises from the fact that the steam is now at a temperature above that of formation. The greater the degree of superheat, the more nearly does this portion of the curve agree with the hyperbolic curve of a perfect gas, expanding isothermally. If a series of such curves be plotted as a result of observations upon the properties of steam, there comes a point when the isothermal ceases to have any horizontal portion. We have now a continuous curve, indicating that for this temperature, and any other above it, no pressure however great will cause the vapour to liquefy.

At the summit of the curve x, x', x'', y'', y', y the substance is in what is known as the critical state, the isothermal line which touches it being a curve of 'critical temperature.'

The critical temperature for steam is about 689 deg. Fahr., and the pressure at the critical point is 2950 pounds

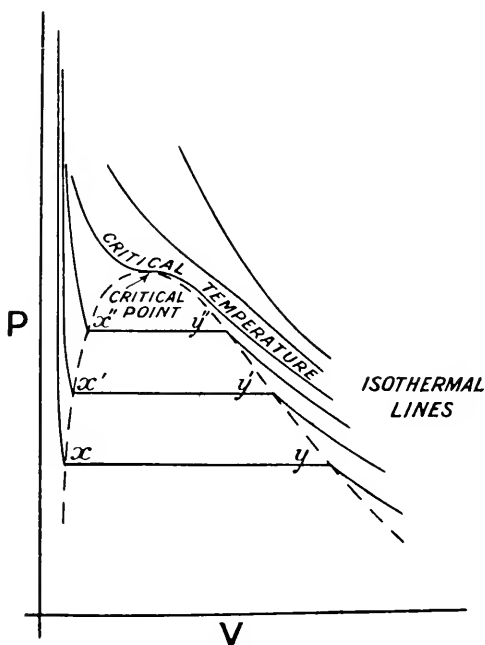


FIG. 12.—Isothermal Curves for Steam.

per square inch. In the region beyond the curve of critical temperature the substance passes into the condition of a permanent gas, and the further removed it is from the critical state the more nearly does it behave as a perfect gas.

Expansion of Steam at Constant Total Heat.—

Imagine the column of steam (i), (fig. 13), whose pressure is P_1 and volume V_1 , to pass through the constricted orifice O into a region where the pressure is P_2 and the volume

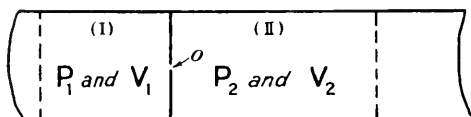


FIG. 13.—Throttling of Steam.

V_2 . Suppose, further, that no heat enters or leaves the vapour during the process. Then in passing from (i) to (ii) work is done upon the steam amounting to P_1V_1/J heat units. In forcing back the vapour in the space (ii) the steam does P_2V_2/J heat units of work. Now since there has been no transfer of heat to the steam from any outside source, if P_2V_2/J exceeds P_1V_1/J , work must have been done at the expense of the internal energy of the steam.

$$\text{Hence} \quad \frac{P_2V_2}{J} - \frac{P_1V_1}{J} = E_1 - E_2$$

$$\text{i.e.} \quad \frac{P_1V_1}{J} + E_1 = \frac{P_2V_2}{J} + E_2$$

or, the total heat is constant during a throttling action.

Adiabatic Expansion of Steam.—If we assume the same general conditions as in the case of expansion at constant total heat, but substitute for the orifice at O a nozzle, say, together with a suitable rotor (or any other device whereby the work done during the expansion of the steam is wholly removed, no part being restored to the steam in the form of heat), we have an adiabatic expansion. This process differs from a throttling action in that the total heat before expansion is greater than the total heat at the end of the expansion by an amount which is exactly equal to the work done.

Referring to fig. 14, the area (i) (ii) (iii) (iv) gives a measure of the work done, which, from the principle of the conservation of energy, is equal to

$$\frac{P_1V_1}{J} - \frac{P_2V_2}{J} + E_1 - E_2$$

$$\text{or} \quad I_1 - I_2.$$

In order to determine the condition of the steam at the end of the adiabatic expansion, and hence to find the drop in total heat, recourse may be had to the temperature-entropy diagram. Whether the steam be wet, dry, or superheated, the entropy remains constant during the adiabatic expansion.

Hence we have the general equation, $\phi_1 = \phi_2$.

Knowing the initial condition of the steam, the final state may easily be found from the steam tables.

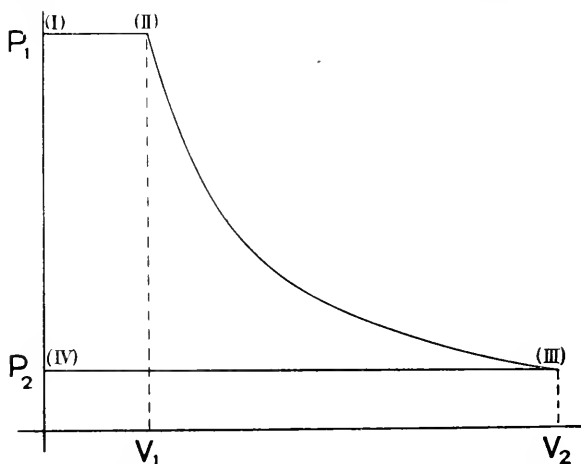


FIG. 14.—Adiabatic Expansion of Steam on PV Plane.

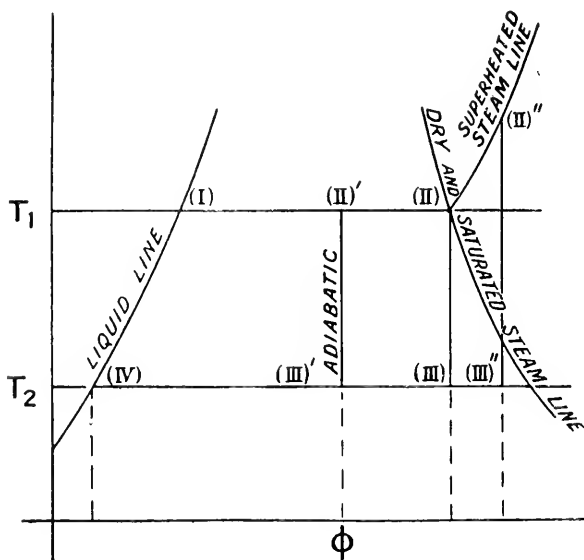


FIG. 15.—Adiabatic Expansion of Steam on $T \phi$ plane.

Another plan which commends itself, on account of its great convenience, is to make use of a chart upon which

are exhibited the properties of steam between co-ordinates of total heat and entropy. This device is known as a Mollier chart.

Referring to the figure, it may be observed that in the wet steam region a series of curves of constant pressure, intersected by a series of curves of constant dryness, are

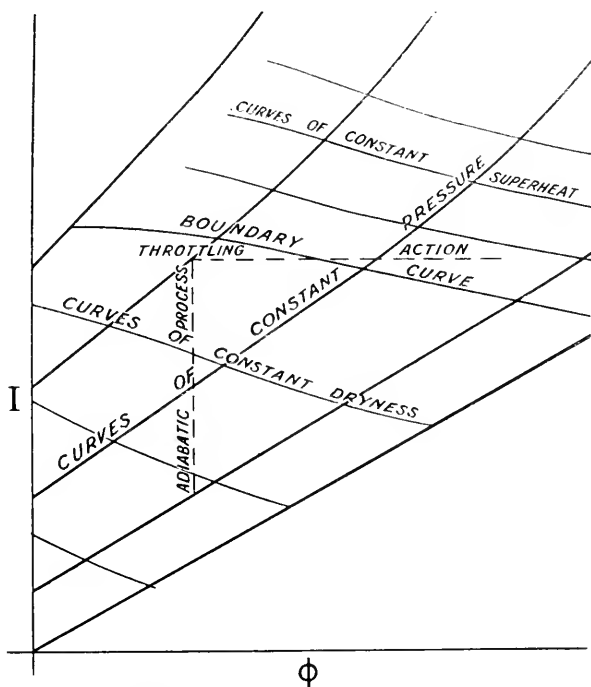


FIG. 16.—Mollier Chart of Total Heat and Entropy.

plotted. In the region of superheat the curves of constant pressure are continued, but are now crossed by curves of constant superheat. The chart serves the purpose of not only giving a ready solution to any problem in which the work done during an adiabatic process is required, but also of indicating with equal facility the condition of steam after a throttling process. In the former case the locus of the state point is a vertical straight line ; in the latter it is horizontal.

Carnot's Cycle, using Steam as the Working Substance.—We may now consider the action of an

engine which is reversible in the Carnot sense, and which uses water and water vapour as the working substance.

Let there be provided, as before, a perfectly non-conducting cylinder and piston which act as a completely insulating envelope enclosing unit mass of the vapour. Let there be available also an infinite source and an infinite receiver of heat, which may be applied at will to the conducting end of the cylinder. This latter may be rendered non-conducting by the use of a cover. The procedure is then similar to that followed in the case of

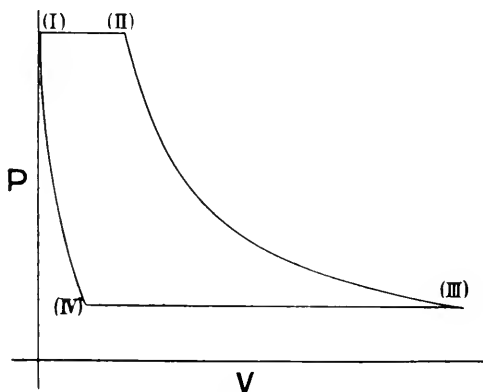


FIG. 17.—Carnot's Cycle with Water and Steam.

a perfect gas. During the first stage we observe that instead of a hyperbolic curve we have a horizontal straight line. This follows from the known properties of steam. Under normal conditions, the transition from unit mass of water at temperature T_1 to dry steam at the same temperature takes place at constant pressure. During the next stage, with the non-conducting cover in position, expansion takes place in such a manner that no heat enters or leaves the steam, this supposition excluding the addition of heat to the working substance by internal actions amongst the parts of the fluid. The condensation which takes place in stage 3 is stopped at such a point that an adiabatic compression restores the mixture of water and water vapour to the condition of water at temperature T_1 . The cycle which we

have thus imagined being reversible has the maximum efficiency, namely, $\frac{T_1 - T_2}{T_1}$.

The form assumed by the diagram of Carnot's cycle, between co-ordinates of temperature and entropy, is (as already indicated) a simple rectangle, and this is true whatever be the nature of the working substance. In the case of steam the rectangle has a height which is proportional to the difference between the temperature limits T_1 and T_2 , and, if the steam be dry at the commencement of the adiabatic expansion in the second stage, the width gives, to some scale, a measure of $\frac{L_1}{T_1}$ where L_1 is the heat of vaporization at T_1 .

Clapeyron's Equation.—This is an interesting relation which exists between the specific volume of a fluid, the heat required to cause change of state (either from the solid to the liquid, or from the liquid to the gaseous state), the temperature at which such change takes place, and the rate of change of temperature with respect to pressure for the particular pressure at which the change takes place. For instance, in the case of steam, imagine a Carnot engine to work between temperature and corresponding pressure limits which are separated by quantities of an infinitesimal order. Then the work done as represented by the pressure-volume diagram is

$$(V_s - V_w) dP$$

(V_s being the volume of unit mass of dry and saturated steam and V_w being the volume of the water from which the steam was generated). From the temperature-entropy diagram we obtain similarly

$$\frac{JL}{T} dT.$$

These quantities are equal,

$$\therefore V_s = \frac{JL}{T} \frac{dT}{dP} + V_w.$$

If, then, a formula can be found which will exhibit the relation between temperature and pressure with

sufficient exactitude, and which may be differentiated, the specific volume of any fluid can be determined when the latent heat is known. It is interesting to observe, for example, that a knowledge of this relationship will convey to an observer, seeing an iceberg float with part of its mass above water, the information that the melting-point of ice is lowered by pressure.

For, since the density of the water is greater than the ice, the volume of the former is less, and hence the ratio $\frac{dT}{dP}$ is negative.

Cycle of the Steam-Engine and Steam-Turbine.—

Owing to mechanical considerations, which will be dis-

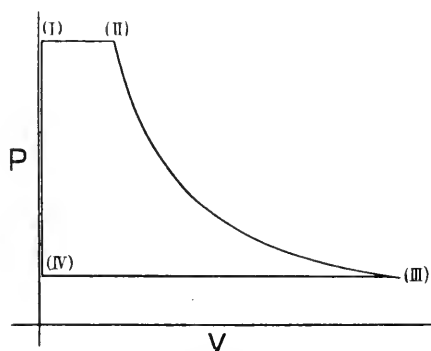


FIG. 18.—Rankine Cycle of the Steam-Engine on PV Plane.

cussed later, the steam engine and turbine, while they follow the Carnot cycle in its first three stages, do not in general complete the action by an adiabatic process. The operations are normally carried out in three separate organs: the boiler, in which heat

is added at constant temperature during the formation of dry steam under constant pressure; the prime-mover, which under ideal conditions would expand the steam adiabatically; and the condenser, which functions as the receiver of heat.

The condensate from the latter in a wholly liquid state is returned to the boiler by means of a pump and is there heated up to the initial temperature, thus restoring the working substance to its original condition.

Since a part of the heat in the Rankine cycle (as this modification of the Carnot is usually termed) is added during the heating of the water from the lower to the upper limit of temperature, and therefore under conditions

such that its availability is less than if it had been added at the top of the range, the efficiency of the cycle as a whole is less than that of Carnot.

Referring to the pressure-volume diagram, we observe that the work done is represented by the area (i) (ii) (iii) (iv). The heat put in is $H_1 - H_2$, where H is equal to $I - PV_r/J$. This gives us, for the exact expression of the efficiency :

$$\frac{I_1 - I_2}{I_1 - I_{w_2}} = \frac{(P_1 - P_2) V_{w_2}/J}{(P_1 - P_2) V_{w_2}/J}.$$

The term $(P_1 - P_2) V_{w_2}$ takes account of the work spent by the feed-pump in introducing the water into the boiler, and which has to be provided by the engine. It is a quantity of small consequence practically and may, in general, be neglected.

If the steam is dry at the commencement of the adiabatic expansion, the expression for the efficiency may be written

$$\frac{I_{s1} - I_2}{I_{s1} - I_{u2}}.$$

Upon the same principle the efficiency under any other conditions, whether the steam be dry, wet, or superheated at the end of the first stage, may be easily determined, when the adiabatic expansion is complete.

If expansion is incomplete it is easier to treat the problem in two parts, the work done in the upper part of the range down to the pressure level at which the expansion is stopped being found as above described, and the balance being calculated from the pressure-volume diagram.

Fig. 19 illustrates the various forms assumed by the

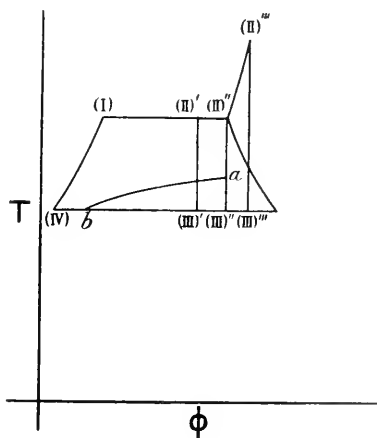


FIG. 19.—Rankine Cycle of the
Steam-Engine on $T \phi$ Plane.

Rankine cycle between co-ordinates of temperature and entropy, when the initial conditions of the adiabatic expansion are different. The case in which expansion is incomplete is also shown, the curve *ab* being plotted from the consideration that the volume of the water-steam mixture remains constant while the pressure and temperature fall. It is to be noted that the area of the temperature-entropy diagram does not give a true measure of the net work done by the engine, since a small part of the gross work available within the cylinder of the engine is spent in doing work upon the feed-pump. As already mentioned, however, this quantity is commonly neglected for practical purposes of comparison of the performance of steam engines and turbines with the ideal standard.

Reversal of the Vapour Engine.—As in the case of an engine using a gaseous working substance, the reversed vapour engine may be used as a cooling machine. A liquid such as anhydrous ammonia, whose boiling-point under suitable working pressure is comparatively low, is usually selected. The action then consists in allowing the cooling medium to evaporate in a series of coils either placed directly in the room to be cooled, or, alternatively, immersed in a brine solution, which may then be circulated as desired. The vapour is then compressed in a pump from the lower limit of pressure to the upper. The boiling-point rising with the pressure, condensation may now be effected in the neighbourhood of the temperature of the available supply of cooling water. The last stage, corresponding to the adiabatic compression of the Carnot cycle and to the heating up of the water in the Rankine, is usually an expansion at constant total heat, the liquid being allowed to stream through a reducing valve in quantities which are proportioned to the demand. For obvious reasons a reversal of the last stage of the Rankine cycle cannot be carried out, and the complications which an attempt to reproduce the reversed Carnot cycle would introduce render a modification desirable. The effect of the expansion valve is not only to cause the loss of the work which would be recovered by the use of an expansion cylinder, but to render the

vapour drier than it would have been at the end of an adiabatic process. This is clearly shown on the temperature-entropy diagram (fig. 20). The curve (i) (iv) may be plotted from the consideration that the total heat remains constant.

In estimating the performance of refrigerating machines it is customary to make reference to the modified standard, the heat removed from the cool body being expressed as a ratio to the heat equivalent of the work spent in its removal. This latter quantity is called the coefficient of performance.

In the temperature-entropy diagram the heat removed is the area (iv) (iii) (vii) (vi), and the heat equivalent of the work is the area (i) (ii) (iii) (v), as may be observed from a consideration of the pressure-volume diagram for the pump, (the term $(P_1 - P_2) V_{liq}/J$ being neglected).

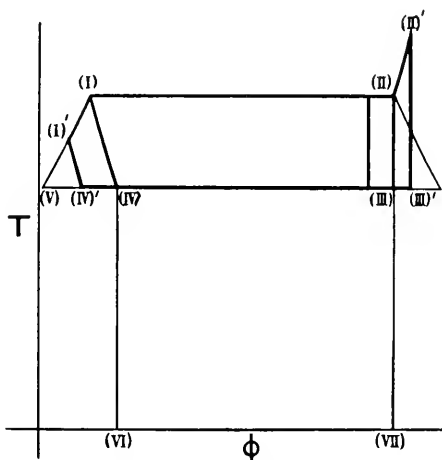


FIG. 20.—Cycle of Refrigerating Machine using a liquid and its vapour.

In practice, in order to increase the refrigerative effect, the liquid, after condensation, is frequently cooled below the upper limit of temperature T_1 in a separate cooler.

A certain amount of superheat at the end of compression also may be tolerated. The influence of these changes upon the temperature-entropy diagram is shown in the figure.

As already indicated in connection with the discussion of a refrigerating machine having a gaseous working substance, the reversed Carnot cycle has the greatest attainable coefficient of performance, between assigned temperature limits.

Any departure from this standard causes a reduction in the value of the coefficient.

In those cases in which the compression is adiabatic, this latter quantity may be determined from a consideration of the changes in total heat, thus :

$$\text{Ideal coefficient} = \frac{I_{(iii)} - I_{(iv)}}{I_{(ii)} - I_{(iii)}}.$$

These quantities may be obtained by calculation from tables of the properties of the working substance, or more readily from the Mollier chart of pressure and total heat.

FLUIDS IN MOTION

FROM our knowledge of the nature of heat and its effect upon matter, notably gases and vapours, methods have been outlined whereby a part of the heat added to the working substance of an engine may be transformed into mechanical work by direct action upon the piston. We have now to consider that case in which the initial transformation is not into directed motion of the piston and its associated linkage, but into ordered motion of the working substance itself. In the ideal case the whole of the heat which disappears in the process reappears as kinetic energy of the moving fluid. This investigation is of the greatest practical consequence, embracing as

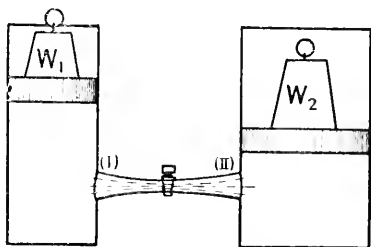


FIG. 21.—Flow through a Channel.

it does the case of the flow of steam and of gas through turbine nozzles and channels, and also the flow of fluids in lines of pipe.

Flow in Channels of Simple Form.—Consider the flow of a fluid through the channel (i) (ii), (fig. 21).

Let it be supposed to move in stream lines and to fill the channel completely. Then as a consequence of these hypotheses, the mass passing an imaginary surface taken at right angles to the stream lines at (i) is equal to the mass passing a similar surface at (ii) in the same time. Imagine, further, that no heat is interchanged between the fluid and its surroundings during the process. Then

by the principle of the conservation of energy, the sum of the energy possessed by the substance itself, together with the work which has been given up by the system with which it was associated at (i), must be equal to the total energy at (ii). In order to illuminate this conception the channel has been shown as a connection between two cylinders whose pistons are loaded with different weights. The passage of a quantity of the fluid from one cylinder to the other causes one piston to rise and the other to fall, the positive work done in one case being P_1V_1 , and the negative work done in the other, P_2V_2 . If, now, E_1 is the internal energy per unit mass at (i), and if E_2 is the internal energy at (ii), then if no chemical or other change takes place,

$$P_1V_1 + E_1 + \frac{v_1^2}{2g} = P_2V_2 + E_2 + \frac{v_2^2}{2g} \text{ (in work units)}$$

if the channel is horizontal, or if the change in potential energy is negligible. If w is the density of the fluid ($= \frac{I}{V}$) the equation may also be written :

$$\frac{P_1}{w_1} + E_1 + \frac{v_1^2}{2g} = \frac{P_2}{w_2} + E_2 + \frac{v_2^2}{2g} \text{ (in work units)}$$

when its identity with that of Bernoulli applied to the flow of water becomes apparent. The term which takes account of the internal energy is commonly omitted in the case of water, while that which represents the potential energy cannot, in general, be neglected.

Again, rewriting the above equation we have

$$\frac{v_2^2 - v_1^2}{2g} = (P_1V_1 + E_1) - (P_2V_2 + E_2) = I_1 - I_2.$$

In other words, the change in kinetic energy is equal to the drop in total heat, all losses by friction, etc. being neglected. It is also instructive to consider this problem from the point of view of the pressure-volume diagram as shown in fig. 22.

The area under (i) (ii) is the work done upon the fluid as it enters the channel. The area under (ii) (iii) is that done by the fluid during the adiabatic expansion in the channel: while the area under (iii) (iv) represents the work done against the back pressure in the region into which the fluid is discharged.

If the law of the adiabatic expansion is known, then the net area representing the work done in causing change of kinetic energy may be determined. For instance,

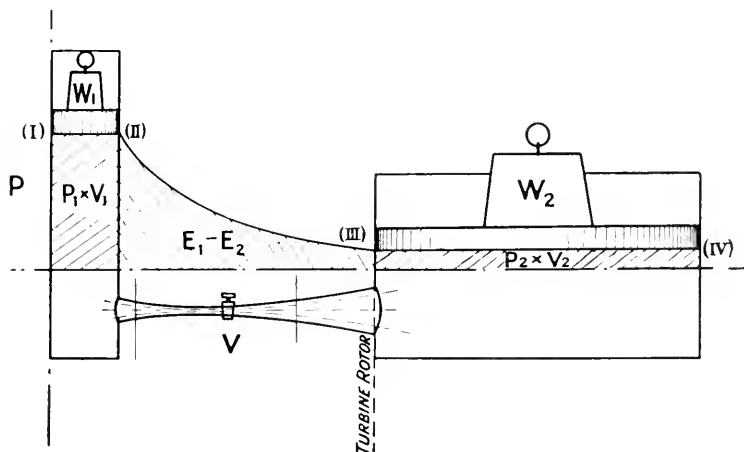


FIG. 22.—Pressure-Volume Diagram during Flow through a Channel.

suppose that the substance under consideration is a perfect gas, then, per unit mass,

$$\begin{aligned} \frac{v_2^2 - v_1^2}{2g} &= P_1 V_1 - P_2 V_2 + \frac{(P_1 V_1 - P_2 V_2)}{\gamma - 1} \\ &= \frac{\gamma}{\gamma - 1} (P_1 V_1 - P_2 V_2) = \frac{C_p}{C_p - 1} (RT_1 - RT_2) \\ &= \frac{C_p R}{C_p - C_v} (T_1 - T_2) = C_p (T_1 - T_2). \end{aligned}$$

In the case of steam the appropriate value of the index of V in the equation PV^n equals a constant may

be selected when the condition of the steam at the commencement of the adiabatic expansion is known. In the region of superheat the figure 1.3 may be taken, while in the wet region n can be calculated from the formula of Zeuner, namely, $n = 1.035 + 0.1q$, q being the dryness of the steam at the commencement of the adiabatic expansion.

The equation for steam then becomes

$$\frac{v_2^2 - v_1^2}{2g} = \frac{n}{n-1} (P_1 V_1 - P_2 V_2),$$

V_1 and V_2 representing, of course, the actual initial and final volumes per unit mass.

If, now, v_1 is negligible, as it commonly is in practice,

$$\begin{aligned} v_2 &= \left\{ \frac{2gn}{n-1} (P_1 V_1 - P_2 V_2) \right\}^{\frac{1}{2}} \\ &= \left\{ \frac{2gn}{n-1} \left[1 - \left(\frac{P_2}{P_1} \right)^{1-\frac{1}{n}} \right] P_1 V_1 \right\}^{\frac{1}{2}}, \end{aligned}$$

for isentropic flow.

This equation is, obviously, of general application, and may be used to determine the velocity at any point in the stream where the pressure P_2 is known.

Let, now, A be the area of the cross-section through which unit mass flows per second when the pressure has fallen to P .

Then

$$v = \frac{V}{A} = \frac{V_1 P_1^{\frac{1}{n}}}{A P^{\frac{1}{n}}}.$$

$$\therefore \frac{1}{A} = \left(\frac{P}{P_1} \right)^{\frac{1}{n}} \frac{1}{V_1} \left\{ \frac{2gn}{n-1} \left[1 - \left(\frac{P}{P_1} \right)^{1-\frac{1}{n}} \right] P_1 V_1 \right\}^{\frac{1}{2}}.$$

Since unit mass is discharged through A per second, therefore $\frac{1}{A}$ gives a measure of the mass of the discharge per unit area per second. At the minimum cross-section

of the channel, the discharge per square foot per second is a maximum.

Let $\frac{P}{P_1}$ be called D . Then

$$\frac{1}{A} = \left(\frac{P_1}{V_1} \frac{2gn}{n-1} \right)^{\frac{1}{2}} \left\{ D^{\frac{2}{n}} - D^{\frac{n+1}{n}} \right\}^{\frac{1}{2}}.$$

When the rate of change of this quantity with respect to the ratio D in which the pressure falls is zero, $\frac{1}{A}$ has its maximum value.

This is when

$$\frac{d \left(D^{\frac{2}{n}} - D^{\frac{n+1}{n}} \right)}{dD} = 0,$$

or when
$$\frac{2}{n} D^{\frac{2}{n}-1} - \frac{n+1}{n} D^{\frac{n+1}{n}-1} = 0,$$

i.e. when
$$D^{\frac{2-n}{n}-\frac{1}{n}} = \frac{n+1}{2},$$

i.e. when
$$D = \left(\frac{n+1}{2} \right)^{\frac{n}{1-n}} = \left(\frac{2}{n+1} \right)^{\frac{n}{n+1}}.$$

In the case of a gas for which $n = \gamma = 1.4$, D assumes the value 0.528. This means that when the back pressure P_2 is less than 0.528 P_1 the channel will consist of a convergent and a divergent portion, and the pressure at the minimum section or throat will be 0.528 times the initial pressure. When the back pressure is greater than 0.528 times the initial pressure, the appropriate channel would be convergent only, and the mass of the discharge would be calculated by substituting P_2 for P_t , the pressure at the throat. Thus if A_{ex} is the area of the exit of a convergent nozzle in square feet, the mass of the discharge in pounds per second is

$$A_{ex} \left\{ \frac{P_1}{V_1} \frac{2gn}{n-1} \right\}^{\frac{1}{2}} \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1} \right)^{\frac{n+1}{n}} \right\}^{\frac{1}{2}}.$$

The values of D for any other fluid may easily be found similarly when n is known with sufficient exactitude.

Thus for steam, when $n = 1.3$, $D = 0.5457$, and when $n = 1.135$, $D = 0.577$.

Metastable State of Steam.—In the above discussion we have tacitly assumed that the behaviour of steam expanding in a nozzle is unaffected by the rapidity of the process and other factors rendering the conditions different from those of the comparatively slow expansion in the cylinder of an engine. It is, however, necessary to extend our observations upon the properties of water and water vapour, in order to account for certain discrepancies which make their appearance when the results of experiments upon the flow of steam are compared with calculations based upon our knowledge of the normal or equilibrium state.

The particles of a gas are largely unaffected by intermolecular attractions and repulsions. As the gas is cooled to the condition of a superheated vapour, there is an increasing tendency towards coaggregation of the molecules into groups.

In the liquid state coaggregation is complete and powerful intermolecular forces prevent closer approach, while forces of much less magnitude tend to resist molecular separation.

Throughout the mass of the liquid these latter forces are internally balanced, but at the surface of separation between the liquid and its enclosing envelope they become manifest, and are responsible for the phenomena associated with surface tension. Thus a drop of water resting upon a surface which it does not wet tends to assume a spherical form, the sum of the potential energy of the drop due to the height of its centre of gravity above the plane of support and the energy stored as surface tension, having a minimum value. If, then, a drop of water evaporate, less heat will have to be supplied than if it were part of the main mass of the fluid, since as its size diminishes the energy given up assists in the formation of vapour. Thus as the radius of the drop decreases, the ratio of potential energy stored in the surface layer to the mass of the drop increases and the tendency to evaporate becomes greater. Hence the pressure at which evapora-

tion will just balance condensation is greater for a drop; or, to put the matter slightly differently, evaporation will continue from the drop after the liquid, at the same temperature, has reached a state of equilibrium. For the same reason the tendency towards condensation in a vapour at the saturation point is resisted, withdrawal of heat resulting in a more or less well-marked cooling below saturation temperature, before condensation commences. A vapour in this condition is said to be supersaturated. If foreign particles are present these will act as nuclei in the first instance, while if these are absent, or present in insufficient numbers, droplets will ultimately collect as a fine mist about nuclei of molecular dimensions. Should the drop, however, carry a static electric charge, the potential rises as the drop becomes smaller. This opposes the tendency towards evaporation.

These considerations afford an explanation also of the retardation of boiling in water which has been largely freed of the tiny air bubbles which are usually present. Evaporation continues from the free surface until the temperature has risen sufficiently to cause the pressure within the exceedingly small bubbles which remain to exceed that without by a quantity which depends upon the size of these gaseous nuclei and the particular value of the surface tension of the water.

Flow of Supersaturated Steam.—As already stated, superheated steam expands very approximately according to the law $PV^{1.3} = \text{a constant}$. If, on adiabatic expansion, supersaturation supervenes, the law holds until reversion to the equilibrium state occurs. How far this metastable condition persists has been the subject of inquiry.

It is known to obtain at least as far as the throat of the turbine nozzle, and probably reversion is incomplete when the steam leaves the exit end of the channel. The influence of supersaturation on the discharge may be seen by inserting the appropriate values of D and n , namely, 0.5457 and 1.3, in the equation for the mass flowing per unit time, and comparing the result so derived with that obtained on the assumption of stable expansion.

The former quantity will be found to be about 5 per cent. in excess of the latter, and gives results which are in accord with experiment when a reasonable allowance is made for friction and other losses.

In order to determine the heat drop of supersaturated steam, recourse may be had to the pressure-volume diagram.

The law of the expansion being known, the total work done between any assigned limits of pressure may be found. Since the action is ideally isentropic, this quantity is a measure of the heat which has disappeared in the process. Further, the temperature of the supercooled

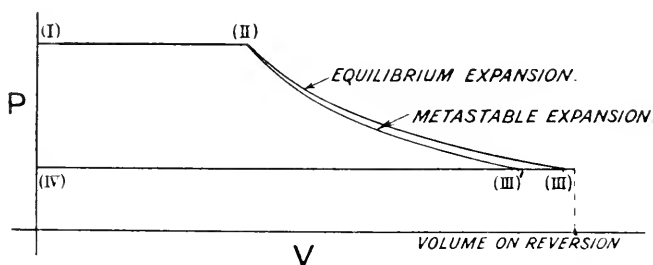


FIG. 23.—Expansion of Supersaturated Steam on PV Plane.

steam may be calculated on the assumption that it behaves as a cool gas. If these calculations are made it will be found that the curve of the adiabatic expansion of supersaturated steam lies below that for the stable state. Hence the work done in the former case is less, the total heat at the end of the process being greater than in the latter. On reversion at constant total heat the steam becomes drier than for equilibrium expansion, as may well be seen from a Mollier chart, which exhibits the properties of both saturated and supersaturated steam.

The influence of supersaturation is similar to the effects of friction in that it produces irreversible actions, causing growth of entropy and loss of availability of the heat energy of the steam.

In fig. 24, (ii) (iii) represents the isentropic expansion of the supersaturated steam ; (iv) is the state

point at the end of the expansion on the assumption of stable conditions; (iii)' indicates the condition of the steam as to dryness on reversion from the supersaturated state. The chain lines are lines of constant pressure for supersaturated steam in the normally wet region.

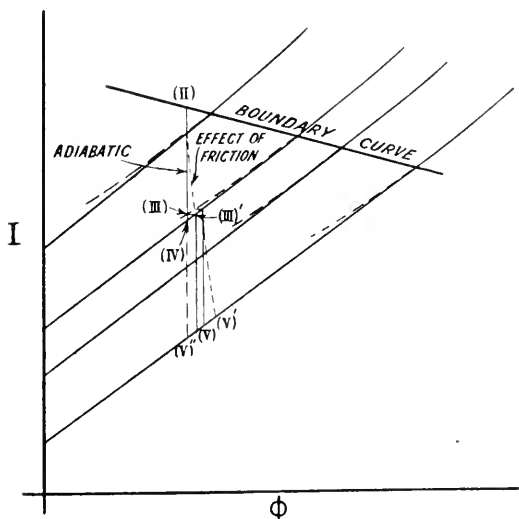


FIG. 24.—Expansion of Supersaturated Steam on $I \phi$ Plane.

Flow in Cylindrical Channels, Friction Considered.—Consider a differential length of pipe line in which a given gas or vapour is flowing.

We have the total energy equation, per unit mass,

$$\frac{P_1}{w_1} + E_1 + \frac{v_1^2}{2g} = \frac{P_2}{w_2} + E_2 + \frac{v_2^2}{2g} + Z,$$

Z being equal to the frictional loss.

If v may be regarded as constant for the segment, and if the heat generated by friction be supposed to return to the fluid, the total heat remains constant.

Then if the fluid be a gas, $E_1 = E_2$ and $\frac{P_2}{w_2} - \frac{P_1}{w_1} = -Z$.

Let, now, the total work lost by an element of the fluid which fills the segment be proportional to the mean velocity of flow (v) and to the density of the fluid (w).

Then total work lost $= f v^2 w (\pi d dl) dl$.

Per unit mass, this $= \frac{f v^2 w (\pi d dl) dl}{\pi d^2 dl w}$.

f being a constant relating to unit velocity and unit area of the inside of the pipe.

$$\therefore -Z = \frac{f' v^2 dl}{d}.$$

If dP is the pressure drop between (i) and (ii),

$$- \frac{dP}{w} = -dPV = \frac{f' v^2 dl}{d}.$$

Now $V = \frac{RT}{P}$ and $v = \frac{V_w}{A} = \frac{WRT}{PA}$.

$$\therefore -dP = \frac{P}{RT} \frac{f'}{d} \left(\frac{WRT}{PA} \right)^2 dl,$$

and $-\frac{dP}{P} = \frac{f' W^2 dl}{d^5}$ if T may be regarded as constant, and equal, say, to 520 deg. Fahr. abs.

On integration we have

$$\frac{P_1^2 - P_2^2}{2} = \frac{f' W^2 L}{d^5}$$

or
$$P_1^2 - P_2^2 = \frac{f'' W^2 L}{d^5}$$

or, if the volume of air, at atmospheric pressure, flowing per second be known, instead of W we may write

$$\frac{P_a V_w}{RT} \text{ and } P_1^2 - P_2^2 = \frac{f''' V_w^2 L}{d^5}.$$

If V_w is in cubic feet, L is in feet, P is in pounds per square inch, and d in inches, f''' may be taken as $\frac{1}{2000}$ for air. Then

$$p_1^2 - p_2^2 = \frac{V_w^2 L}{2000 d^5}.$$

TOPICS FOR DISCUSSION

1. From the simple assumptions made in regard to the properties of a perfect gas, deduce an expression for the pressure, in terms of the mass of the individual molecule, the 'velocity of mean square,' and the number of molecules per unit volume.

2. What is the thermodynamic interpretation of the term 'temperature'?

3. Show that the laws of Boyle, Dalton, and Avogadro, relating to a perfect gas, are a direct consequence of the equations derived from the application of the kinetic theory of heat to a gas.

4. Show that the translational energy of the molecules of unit mass of a gas is equal to $\frac{3}{2} \cdot RT$, and hence determine the value of the ratio of the specific heats (a) of a monatomic gas, and (b) of a diatomic gas.

5. Prove that during the adiabatic expansion of a perfect gas a law of the form $PV^\gamma = \text{constant}$ is followed, γ being equal to a constant, the ratio of the specific heats of the gas.

6. Outline a reversible cyclic process which aims at the conversion of heat into mechanical work. Does a non-cyclic process, such as the infinite isothermal expansion of air, drawing heat, say, from the ocean, constitute a violation of the second law? If not, why not?

7. Show that the scale of the perfect gas thermometer is coincident with Lord Kelvin's scale.

8. Determine the expression for the efficiency of:

(a) The constant pressure internal combustion engine cycle;

(b) The constant volume internal combustion engine cycle.

9. Explain how a machine operating on the reversed Carnot cycle might be employed to either warm or cool a room.

10. What is meant by the total heat of a substance, such as a gas or steam. By how much does this quantity differ from the heat actually added to produce unit mass of dry steam at a given temperature and pressure?

11. Show that the total heat of steam or any other substance is constant during a throttling action.

12. If the melting-point of ice were raised instead of lowered by increase of pressure, show that we might imagine an engine which would violate the laws of thermodynamics.

13. Prove that the work done on the Rankine cycle for steam is equal to the drop in total heat, as found, say, by measuring the length of the isentropic curve on the total heat-entropy chart. In order to apply this method to the Carnot cycle, what further calculation would have to be made?

14. If the reversed Rankine cycle could be employed in a cooling machine instead of the reversed Carnot, would this increase or diminish the coefficient of performance?

15. Verify the figure given for the percentage increase in the discharge of supersaturated steam over that for normal wet expansion, the steam being originally dry and saturated and expanding in a nozzle having a throat.

16. How might a curve of constant pressure for super-saturated steam be drawn in the normally wet region of the total heat-entropy chart?

WORKED EXAMPLES

1. Calculate the 'velocity of mean square' (*a*) for oxygen, (*b*) for nitrogen, under a pressure of 14.7 pounds per square inch and at a temperature of 492 deg. Fahr. abs.

The density of oxygen under the conditions given is

0.0892 pound per cubic foot, and the density of nitrogen under the conditions given is 0.0783 pound per cubic foot.

We have

$$PV = \frac{\frac{1}{3} v^2}{g}$$

$$\begin{aligned} \therefore v &= \sqrt{3PVg} \\ &= \sqrt{\frac{14.7 \times 144 \times 32.2 \times 3}{0.0892}} \\ &= 1514 \text{ feet per second for oxygen.} \end{aligned}$$

Similarly

$$\begin{aligned} v &= \sqrt{\frac{14.7 \times 144 \times 32.2 \times 3}{0.0783}} \\ &= 1616 \text{ feet per second for nitrogen.} \end{aligned}$$

2. Find the net work done if 40 cubic feet of air expand adiabatically from a pressure of 120 pounds per square inch to 30 pounds per square inch against a back pressure equal to that of the atmosphere; γ may be taken as 1.4.

The work done is represented by the area of the figure (i) (ii) (iii) (iv).

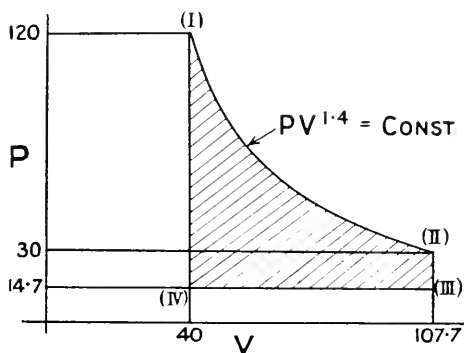


FIG. 25.

$$P_1 = 120 \times 144 \text{ pounds per square foot.}$$

$$P_2 = 30 \times 144 \text{ pounds per square foot.}$$

Since

$$P_1 V_1^\gamma = P_2 V_2^\gamma, \quad \frac{P_1}{P_2} = \left(\frac{V_2}{V_1} \right)^\gamma \quad \text{or} \quad V_2 = V_1 \left(\frac{P_1}{P_2} \right)^{\frac{1}{\gamma}}$$

$$V_2 = 40 \times \left\{ \frac{120 \times 144}{30 \times 144} \right\}^{\frac{1}{1.4}} = 40 \times (4)^{\frac{1}{1.4}} = 107.7 \text{ cu. ft.}$$

Net work

$$\begin{aligned} &= \int_{V_1}^{V_2} P dV - P_a (V_2 - V_1) \\ &= \frac{P_1 V_1 - P_2 V_2}{\gamma - 1} - P_a (V_2 - V_1) \\ &= \frac{144 \times 120 \times 40 - 30 \times 144 \times 107.7}{0.4} - 14.7 \times 144 \times 67.7 \\ &= 565,000 - 143,200 \\ &= \underline{421,800 \text{ foot-pounds.}} \end{aligned}$$

3. A quantity of a certain gas is compressed from initial conditions of 3 cubic feet and 15 pounds per square inch to final conditions of 1.2 cubic feet and 58 pounds per square inch. If the specific heats of the gas at constant volume and constant pressure are 0.173 and 0.244 respectively and the observed rise in temperature of the gas is 294.8 deg. Fahr., calculate the change in internal energy. [B.Sc. (Eng.) Lond.]

The change in internal energy

$$\begin{aligned} &= W C_v (T_2 - T_1) = \frac{C_v (P_2 V_2 - P_1 V_1)}{R} \\ &= \frac{C_v (P_2 V_2 - P_1 V_1)}{C_p - C_v} = \frac{P_2 V_2 - P_1 V_1}{\gamma - 1} \\ &= \frac{144 (58 \times 1.2 - 15 \times 3)}{\frac{0.244}{0.173} - 1} \\ &= \underline{11.09 \text{ B.Th.U.}} \end{aligned}$$

4. The density of CO_2 at 32 deg. Fahr. and 14.7 pounds per square inch pressure is 0.124 pound per cubic foot, and its specific heat at constant pressure is 0.216. A quantity of this gas is compressed adiabatically from a condition of 59 deg. Fahr. and 13 pounds per square inch to a pressure of 50 pounds per square inch. What is the final temperature? [B.Sc. (Eng.) Lond.]

We have $PV = WRT$, W being the mass of the gas under consideration.

We are given that

$$P = 14.7 \times 144 = 2116.8 \text{ pounds per square foot}$$

$$V = \frac{1}{0.124} \text{ per unit mass} = 8.064 \text{ cubic feet}$$

$$T = 32 + 460 = 492 ;$$

$$\text{Hence } R = \frac{2116.8 \times 8.064}{492} = 34.69.$$

$$\text{Now since } C_p - C_v = R, C_v = 0.216 - \frac{34.69}{778} = 0.1714,$$

$$\text{and thus } \gamma = \frac{C_p}{C_v} = \frac{0.216}{0.1714} = 1.260.$$

When a gas expands or is compressed adiabatically,

$$PV^\gamma = \text{const.}$$

$$\text{Thus we have } P_1 V_1^\gamma = P_2 V_2^\gamma$$

and also

$$P_1 V_1 = RT_1 \text{ per pound}$$

$$P_2 V_2 = RT_2 \quad ,, \quad ,,$$

hence

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{1 - \frac{1}{\gamma}}$$

$$= 519 \left(\frac{50 \times 144}{13 \times 144} \right)^{1 - \frac{1}{1.260}}$$

$$= 686^\circ \text{ Fahr. abs.}$$

$$= \underline{\underline{226^\circ \text{ Fahr.}}}$$

5. The characteristic constant for hydrogen is 768 foot-pound units, and its specific heat at constant pressure is $3\cdot41$. Three cubic feet of hydrogen measured at 15 pounds per square inch and $64\cdot4$ deg. Fahr. are compressed adiabatically to 200 pounds per square inch and then expanded isothermally to the original volume of 3 cubic feet. Determine the final pressure of the gas.

Calculate the amount of heat which must be added to the gas during isothermal expansion and also the heat which must be abstracted from the gas after expansion in order to reduce it to the initial state of pressure. [B.Sc. (Eng.) Lond.]

$$C_v = C_p - \frac{R}{J} = 3\cdot41 - \frac{768}{778} = 2\cdot423;$$

hence
$$\gamma = \frac{3\cdot41}{2\cdot423} = 1\cdot407.$$

We have $P_1 V_1^\gamma = P_2 V_2^\gamma$ and $P_2 V_2 = P_3 V_3$,

$$\begin{aligned} \therefore P_3 &= \frac{P_2 \left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}} V_1}{V_3} \\ &= \frac{200 \times 144 \left(\frac{15}{200}\right)^{\frac{1}{1\cdot407}} \times 3}{3} \\ &= 4566 \text{ pounds per square foot.} \end{aligned}$$

Final pressure = $31\cdot7$ pounds per square inch . . . (i)

The heat added to the gas during isothermal expansion is the exact equivalent of the work done, since the temperature and hence the internal energy remain constant ;

$$\begin{aligned} \text{i.e. heat added} &= \int_{V_2}^{V_3} P dV \\ &= \frac{P_2 V_2}{J} \log_e \frac{V_3}{V_2}. \end{aligned}$$

$$\text{Now, } V_2 = \frac{P_3}{P_2} V_3 = \frac{31.7}{200} \times 3 = 0.4755 \text{ c.f.}$$

$$\begin{aligned} \text{Heat added} &= \frac{200 \times 144 \times 0.4755 \times 2.303 \times 0.7999}{778} \\ &= 32.43 \text{ B.Th.U.} \quad \dots \dots \dots \text{(ii)} \end{aligned}$$

In order to find the heat which must be abstracted from the gas to restore it to its original pressure we must find $T_2 = T_3$.

$$\begin{aligned} T_2 &= T_1 \left(\frac{P_2}{P_1} \right)^{\frac{1}{\gamma}} \\ &= 524.4 \times (13.33)^{1-0.711} \\ &= 1109^\circ \text{ Fahr. abs.} \end{aligned}$$

We now have heat abstracted $= C_r W (T_2 - T_1)$,
'W' being the mass of the gas concerned.

From the fundamental equation,

$$W = \frac{PV}{RT} = \frac{15 \times 144 \times 3}{768 \times 524.4}$$

$$\begin{aligned} \text{and heat abstracted} &= \frac{2.423 \times 15 \times 144 \times 3 \times 584.6}{768 \times 524.4} \\ &= 22.79 \text{ B.Th.U.} \quad \dots \dots \dots \text{(iii)} \end{aligned}$$

This latter figure should obviously agree with the heat equivalent of the work spent upon the gas during adiabatic compression, namely,

$$\frac{P_1 V_1 - P_2 V_2}{(\gamma - 1) J} = \frac{15 \times 144 \times 3 - 200 \times 144 \times 0.4755}{(1.407 - 1) 778}$$

which gives 22.8 B.Th.U.

6. Determine the change in entropy which occurs when unit mass of a given mixture of gases passes through the constant volume cycle from initial conditions of pressure 14 pounds per square inch and temperature 600 deg. Fahr. abs., if the ratio of adiabatic compression

is 4, and the maximum pressure reached is 300 pounds per square inch.

C_v for the purposes of this example may be taken as constant and equal to 0.17 , and γ as 1.4 .

We have, for the adiabatic curves, the relations $P_1V_1^\gamma = P_2V_2^\gamma$ and $P_4V_4^\gamma = P_3V_3^\gamma$; also the fundamental equation $PV = RT$ per unit mass.

If we consider the temperature-entropy diagram for this cycle, we observe that the positive change of entropy during the heating process from (iv) to (i) is numerically equal to the negative change during the cooling at constant volume from (ii) to (iii). See fig. 8.

Hence let us determine T_4 and T_1 .

We have $P_3V_3^\gamma = P_4V_4^\gamma$ and $\frac{P_3V_3}{P_4V_4} = \frac{T_3}{T_4}$.

$$\therefore T_4 = T_3 \frac{P_4V_4}{P_3V_3} = T_3 \left(\frac{V_3}{V_4}\right)^\gamma \frac{V_4}{V_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} T_3.$$

$$\therefore T_4 = (4)^{0.4} T_3 = 1045^\circ \text{ Fahr. abs.}$$

Again, we have $P_1V_1 = RT_1$ and $P_3V_3 = RT_3$.

$$\therefore T_1 = T_3 \frac{P_1V_1}{P_3V_3} = 600 \times \frac{300 \times 1}{14 \times 4} = 3214^\circ \text{ Fahr. abs.}$$

The change of entropy = $C_v \int_{T_4}^{T_1} \frac{dT}{T}$

$$= C_v \log_e \frac{T_1}{T_4} = 0.17 \times 2.3026 \times \log \frac{3214}{1045} = 0.1909.$$

This, as above explained, is the same as the change from (iii) to (ii).

7. A quantity of air having a volume of 2 cubic feet at atmospheric conditions of 14.7 pounds per square inch and 59 deg. Fahr. is compressed according to the law

$$PV^{1.15} = \text{a constant}$$

until its pressure is 120 pounds per square inch. Find

the change in internal energy of the air and also the change of entropy. The specific heats of air at constant pressure and constant volume are 0.238 and 0.169. [B.Sc. (Eng.) Lond.]

The change in internal energy = $WC_v (T_2 - T_1)$

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{1-\frac{1}{n}}$$

$$= 519 \left(\frac{120}{14.7} \right)^{0.1304}$$

$$= 682.5^\circ \text{ Fahr. abs.}$$

$$R = (C_p - C_v) J$$

$$= (0.238 - 0.169) 778$$

$$= 53.7$$

$$W = \frac{P_1 V_1}{RT_1}$$

$$= \frac{14.7 \times 144 \times 2}{53.7 \times 519}$$

$$= 0.1519 \text{ pound.}$$

The change in internal energy

$$= 0.1519 \times 0.169 (682.5 - 519)$$

$$= 4.198 \text{ B.Th.U.}$$

For any substance, the change of entropy is measured by

$$\int_1^2 \frac{dQ}{T}$$

When heat is added to a substance it may become manifest in either or both of two ways, namely, as a change in the stock of internal (or intrinsic) energy, or as work done.

In the case of the mass of air under consideration, the former is represented by

$$\int_{T_1}^{T_2} WC_v dT$$

and the latter by $\int_{V_1}^{V_2} \frac{PdV}{J}$

Hence the change of entropy becomes

$$\begin{aligned} & \int_1^2 \frac{WC_v dT}{T} + \int_1^2 \frac{PdV}{JT} \\ &= \int_1^2 \frac{WC_v dT}{T} + \int_1^2 \frac{WRTdV}{JTV} \\ &= W \left\{ C_v \log_{\epsilon} \frac{T_2}{T_1} + \frac{R}{J} \log_{\epsilon} \frac{V_2}{V_1} \right\} \\ &= W \left\{ C_p \log_{\epsilon} \frac{T_2}{T_1} - \frac{R}{J} \log_{\epsilon} \frac{P_2}{P_1} \right\} \\ &= 0.1519 \left\{ 0.238 \times 2.303 \times \log \frac{682.5}{519} - 0.069 \right. \\ & \qquad \qquad \qquad \left. \times 2.303 \times \log \frac{120}{14.7} \right\} \\ &= 0.1519 (0.06518 - 0.1449) \\ &= -0.01212. \end{aligned}$$

8. State and prove the relation existing between the specific heats of a gas at constant pressure and constant volume and the characteristic constant for the gas.

If one-tenth of a pound of gas occupies 1.345 cubic feet at 59 deg. Fahr. and 15 pounds per square inch, and if after adiabatic compression to 0.5 cubic foot the

pressure is 60·9 pounds per square inch, what is the value of the specific heat at constant pressure of this gas? [B.Sc. (Eng.) Lond.]

If a given quantity of a gas be heated under constant pressure P , from T_1 to $T_1 + dT$, expansion takes place according to Charles' law, the external work done being PdV .

If the same quantity of gas be heated from T to $T_1 + dT$ under constant volume, no external work is done, and the heat added goes to increase the internal energy of the gas.

Now we remember that by Joule's law the internal energy of the gas depends wholly on the temperature. Hence the difference between the heat added in the first and in the second case must be a measure of the external work, since the increase in the internal energy is the same for both.

Therefore, per unit mass,

$$(C_p - C_v) dT = \frac{PdV}{J},$$

C_p and C_v being expressed in heat units.

Now when P is a constant,

$$PdV = RdT,$$

Therefore

$$C_p - C_v = \frac{R}{J}.$$

During adiabatic compression the gas follows a law of the form

$$P_1 V_1^\gamma = P_2 V_2^\gamma$$

Therefore

$$\begin{aligned} \log P_1 + \gamma \log V_1 &= \log P_2 + \gamma \log V_2 \\ \therefore \gamma &= \frac{\log P_2 - \log P_1}{\log V_1 - \log V_2} \\ &= \frac{\log (60 \cdot 9 \times 144) - \log (15 \times 144)}{\log 1 \cdot 345 - \log 0 \cdot 5} \end{aligned}$$

$$= \frac{1.7846 - 1.1761}{0.1287 - 1.6990}$$

$$= 1.411.$$

$$\text{Now } R = \frac{PV}{WT}$$

$$= \frac{15 \times 144 \times 1.345}{0.1 \times (59 + 460)}$$

$$= 55.98.$$

Hence we have

$$\frac{C_p}{C_v} = 1.411$$

$$C_p - C_v = 55.98$$

$$C_p - \frac{C_p}{1.411} = 55.98$$

$$C_p = \underline{0.247}.$$

9. Determine the condition of steam after (a) an adiabatic expansion, (b) a throttling action, from initial conditions of pressure 100 pounds per square inch, and dryness 90 per cent. to 3 pounds per square inch.

In order to solve this problem, either the Mollier chart of entropy and total heat may be used, or the tables of the properties of steam may be resorted to.

Adopting the latter method, we have for an adiabatic process $\phi_1 = \phi_2$.

$$\phi_1 = 0.4743 + 0.9 \times 1.1277 = 1.4892$$

$$\phi_2 = 0.2008 + q \times 1.6840$$

$$\therefore q = \frac{1.4892 - 0.2008}{1.6840} = \underline{0.765}.$$

For a throttling action $I_1 = I_2$.

$$\therefore 298.3 + 0.9 \times 888.0 = 109.4 + q \times 1012.3$$

$$q = \frac{1097.5 - 109.4}{1012.3} = \underline{0.976}.$$

The steam is therefore still wet at the end of the process.

10. Find the efficiency of a Carnot cycle, using water and water vapour between temperature limits corresponding to pressures of 200 and 20 pounds per square inch. Compare the result with the efficiency of an engine following the Rankine cycle working between the same pressure limits and using steam of 90 per cent. dryness. What influence has increasing wetness upon the efficiency of the Rankine cycle?

(i) The temperature of saturated steam corresponding to a pressure of 200 pounds per square inch is 381.9 deg. Fahr., and to 20 pounds per square inch 228.0 deg. Fahr.

Hence the Carnot efficiency

$$= \frac{381.9 - 228.0}{381.9 + 459.6} = 18.29 \text{ per cent.}$$

The efficiency of the Rankine cycle is $\frac{I_1 - I_2}{I_1 - I_{w2}}$

$$I_1 = 354.9 + 758.9 = 1113.8.$$

$$I_2 = 196.1 + q \times 960.$$

q may be found from the Mollier chart to be 0.795 .

Whence efficiency

$$= \frac{1113.8 - 959.1}{1113.8 - 191.1} = 16.7 \text{ per cent.}$$

(ii) Increasing wetness diminishes the efficiency, since a larger proportion of the heat is added at temperatures below the maximum.

11. Find the increase in the velocity of the flow of steam between two points in a channel, if at the commencement of the action the steam is dry, and if the initial velocity and pressure are 500 feet per second and 40 pounds per square inch respectively, the final pressure being 20 pounds per square inch.

We have
$$I_1 + \frac{v_1^2}{2gJ} = I_2 + \frac{v_2^2}{2gJ};$$

or
$$\frac{v_2^2}{2gJ} - \frac{v_1^2}{2gJ} = I_1 - I_2.$$

Hence $v_2 = \{2gJ(I_1 - I_2) + v_1^2\}^{\frac{1}{2}}$.

Now, $I_1 - I_2$ is found from the Mollier chart to be 51.7 B.Th.U. ;

$$\begin{aligned}\therefore v_2 &= \{2 \times 32.2 \times 51.7 + 500^2\} \\ &= 1685 \text{ feet per second.}\end{aligned}$$

Thus the increase in the velocity is $1685 - 500 = 1185$ feet per second.

EXAMPLES FOR PRACTICE

1. The capacity of a balloon is 16,000 cubic feet, and its total weight, including the car, etc., is 1000 pounds. If it be inflated with 13,000 cubic feet of hydrogen at a temperature of 60 deg. Fahr. and 14.7 pounds per square inch, determine the weight of the hydrogen and the lifting effort. Find also the pull on the anchor rope if the pressure of the atmosphere falls to 11 pounds per square inch and the temperature to zero Fahr. The density of hydrogen at 32 deg. Fahr. and 14.7 pounds per square inch is 0.0056. The density of air at 32 deg. Fahr. and 14.7 pounds per square inch is 0.0807.

2. If there is 23 per cent. by weight of oxygen in the air, the balance being nitrogen, determine the pressure due to each separately when the sum of the pressures is 14.7 pounds per square inch. The density of oxygen is 0.0892 at 32 deg. Fahr. and 14.7 pounds per square inch. The density of nitrogen is 0.0783 at 32 deg. Fahr. and 14.7 pounds per square inch.

3. Air at a temperature of 60 deg. Fahr. is throttled from a pressure of 50 pounds per square inch to 25 pounds per square inch.

Using the figure already given for the density, find its volume at the end of the throttling process.

4. Find the work done in the compression and delivery of 200 pounds of air, from normal atmospheric pressure and temperature of 60 deg. Fahr. to 80 pounds per square inch, if the compression is adiabatic. ($\gamma = 1.4$.)

If the compression had been isothermal, how much work would have been saved?

5. A Carnot cycle, using air as the working substance, has an expansion ratio of 3, and an upper and lower temperature limit of 300 deg. Fahr. and 80 deg. Fahr. Find the heat taken in and the work done per unit mass of air.

6. A constant pressure cycle using air works between pressure limits of 14 pounds per square inch and 300. If the temperature at the commencement of the adiabatic compression is 300 deg. Fahr., determine the temperature and entropy of the working substance at the commencement and at the end of the expansion at constant pressure. ($R = 53.4$ and $C_v = 0.17$.) The entropy at the commencement of the adiabatic compression may be taken as the datum.

7. A reversed Carnot engine works between temperature limits of 80 deg. Fahr. and zero. If 10 horse-power is applied to this machine, determine the amount of ice which could be produced from water at 40 deg. Fahr. per hour. The latent heat of ice may be taken as 144 B.Th.U.

8. Steam of 110 pounds per square inch is throttled in a throttling calorimeter to atmospheric pressure, when its temperature is found to be 220 deg. Fahr. Determine its condition before throttling.

9. Steam of 150 pounds per square inch and 100 degrees of superheat is expanded adiabatically. Find the pressure at which it becomes dry and saturated.

10. Given the formula

$$\log_{10} p = 8.360 - \frac{5898}{T} - 0.652 \log_{10} T,$$

in which p equals pressure of saturation of the vapour of mercury in pounds per square inch, and T equals the absolute temperature in Fahr. deg., make use of Clapeyron's equation to find an expression for the specific volume of dry and saturated mercury vapour, in terms of the latent heat L , the rate of change of pressure with respect to the temperature of saturation, and the

absolute temperature. If under a pressure of 70 pounds per square inch the temperature is 856.6 deg. Fahr. and latent heat is 118.2 , calculate the specific volume. [B.Sc. (Eng.) Lond.]

11. A steam-engine works upon the Rankine cycle between pressure limits of 250 pounds per square inch and 5 pounds per square inch. If the steam on entry has 100 deg. of superheat, and if expansion proceeds only as far as 30 pounds per square inch, determine the efficiency.

12. A reversed vapour engine, using ammonia, works between temperature limits of 14 deg. Fahr. and 75.2 deg. Fahr. If there is undercooling of the liquid to 60.8 deg. before entering the expansion valve, determine the ideal coefficient of performance.

13. Calculate the velocity of discharge of air from a container in which the pressure is 25 pounds per square inch, through a nozzle into the atmosphere at 14.7 pounds per square inch. Take C_p as 0.17 and γ as 1.4 .

14. A nozzle is required for the discharge of 400 pounds of steam per hour from a pressure of 140 pounds per square inch to that of the atmosphere. Determine the diameter of the throat and of the exit, on the assumption that supersaturation persists as far as the throat, and that thereafter reversion to the stable condition occurs. Neglect all losses due to friction, radiation, etc.

15. Find a suitable size of air main to deliver 5000 cubic feet of air per minute of atmospheric pressure, if the pressure of delivery is 150 pounds per square inch and temperature 60 deg. Fahr., the permissible pressure drop being 1 pound per square inch per 70 feet.

SECTION II

THERMODYNAMICS OF THE BOILER—
THE BINARY VAPOUR PRIME-MOVER
—THEORY OF THE STEAM-ENGINE,
INCLUDING THE UNIFLOW — THE
STEAM-TURBINE

BOILERS

THE boiler must be regarded thermodynamically as part of the organic group, consisting of boiler, prime-mover, and condenser. We are not here intimately concerned with the design and construction of the multitudinous varieties of steam-boiler. Rather are we interested in their performance from a thermodynamic standpoint, and with a view to discovery in what direction an increase in efficiency in relation to the prime-mover is to be looked for.

Temperatures reached on Combustion.—The chemical potential energy which is released during the combustion of coal or oil is capable of conversion into heat of the products of combustion at a temperature which is only limited by the quantity of heat given out per unit mass of combustible, the mass and specific heat of the products of combustion, and the initial temperature of the air

Thus, to take a simple illustration, 1 pound of carbon gives out on burning to CO_2 , 14,540 B.Th.U. Now the molecular weight of carbon is 12, and that of oxygen 16. Hence the weight of air required is $\frac{32}{12} \times \frac{100}{23}$ pounds per pound of carbon = 11.59 pounds. This gives us 12.59 pounds of carbon dioxide + nitrogen.

If the mean specific heat for the range is taken as 0.29, we have

$$t_1 - 60 = \frac{14540}{12.59 \times 0.29}$$

whence t_1 assumes the value 4090 deg. Fahr., approximately.

In practice, air in excess of the theoretical minimum

would be necessary and the temperature would be proportionately reduced.

However, temperatures around 2500 deg. Fahr. are commonly met with in the case of coal, while figures in the neighbourhood of 3000 degrees may be expected in the case of oil.

Cycle of the Furnace Gases and of the Steam.—

Let us think, then, for a moment of the availability of the heat so communicated to the furnace gases in their passage from the outside of the furnace (in the form of fuel and air) to their final condition at high temperature inside the combustion chamber. We have already seen that the maximum availability for conversion into work of the heat possessed by a body depends upon the temperature at which that heat was added, and the temperature at which the balance of the heat may be rejected.

The measure of this availability is $\frac{T_1 - T_2}{T_1}$.

Now, since the furnace gases under ordinary circumstances are at or near atmospheric pressure, anything in the nature of an expansion becomes impossible, and the only way in which a part of the heat energy could be recovered from the hot gases directly would be to cool them at constant volume back to atmospheric temperature, and complete the cycle by an adiabatic or isothermal compression. The temperature-entropy diagram for this process would be as shown, the area enclosed in each case giving a measure of the work done.

Let, however, the gases part with their heat to another body, such as steam under pressure, so that expansion is possible. We should then have a diagram of the character shown in the next figure, which exhibits the cycle followed by the gases in the furnace and by the steam throughout the cycle of events occurring in the boiler, engine, and condenser.

It is apparent from the diagram that, even with the comparatively high steam pressures now in common use, quite the worst feature of the boiler, thermodynamically, is the irreversible drop which occurs between the temperature level of the hot gases on the one side of the boiler

tubes or plating and that of the steam on the other. Further, if the properties of steam throughout the higher ranges of pressure are considered, it may be observed that as the upper limit of pressure is raised, the ratio which obtains between the heat of vaporization and the liquid heat diminishes. Hence the ratio of the efficiency of the Rankine cycle to that of Carnot also diminishes, falling from a figure of about 90 per cent. when the upper

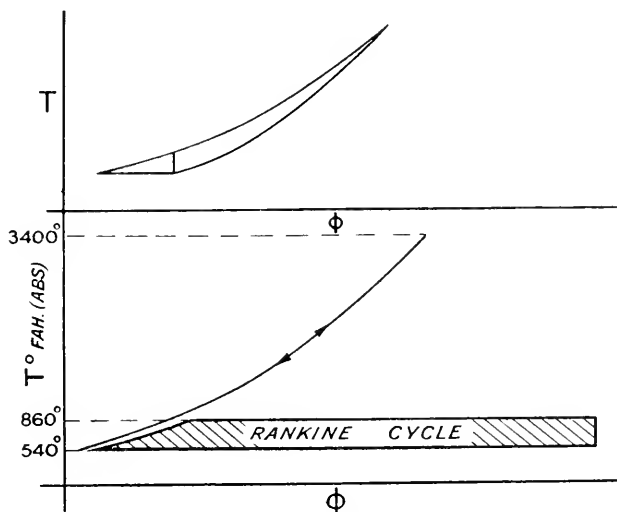


FIG. 26.—Temperature-Entropy Diagram for Furnace Gases.

pressure is 67 pounds per square inch, to 83 per cent. when the upper pressure is 1540 pounds per square inch.

Even supposing it were practicable to use this high pressure, the thermodynamic gain would be altogether disproportionate, the temperature of saturation being only 600 deg. Fahr. The desirability, therefore, of selecting some substance which will vaporize at temperatures which are conveniently much higher than those of steam is apparent.

The Use of Mercury in Conjunction with Steam.—One of the most recent and successful attempts to convert a greater proportion of the energy available in the hot furnace gases into work is that

known as the Emmet Mercury Vapour Process. Under this system mercury is used in conjunction with steam, the mercury being caused to pass through a cycle lying in the upper part of the temperature range, while the steam operates in the lower.

Before proceeding to a discussion of the thermodynamic and practical aspects of this problem, a consideration of the properties of mercury may not come amiss.

¹ **Properties of Mercury and Mercury Vapour.**—

The relation between the vapour pressure of mercury and the corresponding temperature may be taken as

$$\log_{10} p = 8.360 - \frac{5898}{T} - 0.652 \log_{10} T$$

in which

p is in pounds per square inch, and
 T is in degrees Fahr. abs.

The heat of vaporization is given by

$$L = 128.2 - 0.022(T - 860)$$

in which L is the latent heat in B.Th.U.

The specific volume may now be found by the use of Clapeyron's equation :

$$V_m = \frac{JL}{T} \frac{dT}{dP}$$

care being taken in its use to express pressure in pounds per square foot. The value of rate of increase of the temperature with respect to the pressure, for any given pressure, may be found from the equation given above, as follows :

$$\log_{10} p = 8.360 - \frac{5898}{T} - 0.652 \log_{10} T.$$

$$\therefore \frac{\log_e p}{2.303} = 8.360 - \frac{5898}{T} - 0.652 \frac{\log_e T}{2.303}.$$

$$\therefore \log_e p = 2.303 \times 8.360 - \frac{5898 \times 2.303}{T} - 0.652 \log_e T.$$

$$\therefore \frac{dp}{p} = + \frac{5898 \times 2.303 \times dT}{T^2} - 0.652 \frac{dT}{T}.$$

¹ 'The Emmet Mercury Vapour Process,' by W. L. R. Emmet, *Trans. Am. Soc. Mech. Eng.*, 1924.

$$\begin{aligned}\therefore \frac{dp}{dT} &= \frac{5898 \times 2.303 p}{T^2} - 0.652 \frac{p}{T} \\ &= \frac{13580 p}{T^2} - \frac{0.652 p}{T}.\end{aligned}$$

By way of example, let a temperature of 884 deg. Fahr. be selected. It is required to determine the pressure which corresponds to this temperature, the latent heat, and the specific volume.

$$\begin{aligned}\log_{10} p &= 8.360 - \frac{5898}{884 + 460} - 0.652 \log_{10} (884 + 460) \\ &= 8.360 - 4.389 - 2.039 \\ p &= 85.5 \text{ pounds per square inch.}\end{aligned}$$

Again,

$$\begin{aligned}L &= 128.2 - 0.022(T - 860) \\ &= 128.2 - 0.022(1344 - 860) \\ &= 117.56 \text{ B.Th.U.}\end{aligned}$$

$$\text{Also } V_m = \frac{JL}{T} \frac{dT}{dP}$$

$$\begin{aligned}\text{and } \frac{dP}{dT} &= \frac{13580 \times 85.5 \times 144}{1344^2} - \frac{0.652 \times 85.5 \times 144}{1344} \\ &= 86 \text{ nearly.}\end{aligned}$$

$$\begin{aligned}\therefore V_m &= \frac{778 \times 117.56}{1344} \times \frac{1}{86} \\ &= 0.79 \text{ cu. ft.}\end{aligned}$$

The density may then be found from

$$\text{Density} = \frac{1}{V_m}.$$

The specific heat of liquid mercury is 0.033, as far as is known.

The specific heats of mercury vapour, namely C_p and C_v , are determined from the following considerations.

From a knowledge of the atomic weight of mercury ($\text{Hg} = 200$) and of the specific gravity of the vapour, we conclude that the molecules consist of single atoms.

Hence the ratio of the specific heats, according to the kinetic theory, should be 1.667. Again, we know that equal volumes of different gases at the same temperature and pressure contain the same number of molecules, and that under these conditions the communicable energy per molecule is the same for all. This means that the specific heat of different gases is inversely proportional to the molecular weights, since the heat necessary to raise the temperature of approximately equal volumes of different gases under the same conditions is constant.

Thus, if we know R in the equation

$$PV = RT$$

for any gas whose molecular weight is known, then R may be found for any other whose molecular weight is also known.

Suppose we take oxygen, for which R is 48.25. Then R for mercury vapour is

$$\frac{48.25 \times 32}{200} = 7.764,$$

or in heat units

$$\frac{7.764}{778} = 0.00998.$$

Now from the equations

$$\frac{C_p}{C_v} = 1.667 \text{ and } C_p - C_v = 0.00998$$

C_p and C_v may be found to be 0.02482 and 0.01484 respectively.

In calculating the entropy of the liquid the specific heat may be assumed constant and equal to the figure already given, namely, 0.033. The expression then becomes

$$\phi_x - \phi_o = 0.033 \log_e \frac{T_x}{T_o}.$$

The evaporation entropy is found from

$$\phi_{mv} = \frac{L_x}{T_x}.$$

Turbine Cycles for Mercury and Steam.—If the foregoing relations establishing the properties of mercury and mercury vapour be put to use in the construction of tables, it will be found that mercury meets the requirements of a working substance which will operate in the upper region of temperature without the use of excessive pressure. We may now consider, therefore, the thermodynamic advantages to be expected from its use.

Ideally, the more nearly we approach the cycle of Carnot, the higher the efficiency of the prime-mover. Such efficiency, however, must always be considered in its relation to that of the boiler, prime-mover, condenser group. It would, for instance, be possible to imagine an engine operating on the cycle of Carnot between widely separated limits of temperature, which nevertheless converted but a small portion of the heat energy of the fuel supplied to the boiler into work, on account of the excessive waste of heat in the flue gases.

We might, of course, mitigate this latter evil by preheating the air for combustion. If this process were carried to considerable length, however, we should find ourselves in practical difficulties over excessive temperature in the combustion chamber. Practically we are compelled, in general, to adopt a compromise and agree to tolerate a certain reduction in the efficiency of the prime-mover, in order that the boiler efficiency may not be unduly low.

Let us see, first, whether an approach to the Carnot ideal is possible for the mercury in association with a steam-turbine. If the turbine operates upon the Rankine cycle the efficiency, as we know, will be less than the maximum.

A nearer approach to the ideal, however, may be made by a process known as 'bleeding' the turbine. This consists of tapping the cylinder of a compound turbine at a number of points separated by pressure increments, and using the steam so withdrawn to heat the feed progressively. This has the same effect as the regenerative device of the Stirling cycle, and in the limit

the efficiency of such a cycle is that of Carnot, since all the heat supplied from external sources is added at the upper limit of temperature, and all the heat rejected to the condenser leaves at the lower limit. Now, if matters are so arranged that all the heat supplied by the furnace is used in the evaporation of the mercury, and all the heat rejected by the mercury turbine passes into the steam at the temperature of condensation of the mercury (regenerative feed, heating being employed in both prime-movers), we have an ideal arrangement as far as the turbines are concerned, the temperature-entropy diagram being as shown in fig. 27.

In the figure the upper limit of pressure has been taken as 85 pounds per square inch for the mercury, corresponding to a temperature of 884 deg. Fahr., the lower limit as 0.4 pound per square inch, corresponding to a temperature of 402 deg. Fahr.

The upper limit of pressure of the steam is 250 pounds per square inch, corresponding to 401 deg. Fahr., and the lower limit 0.5 pound per square inch, corresponding to 80 deg. Fahr., nearly.

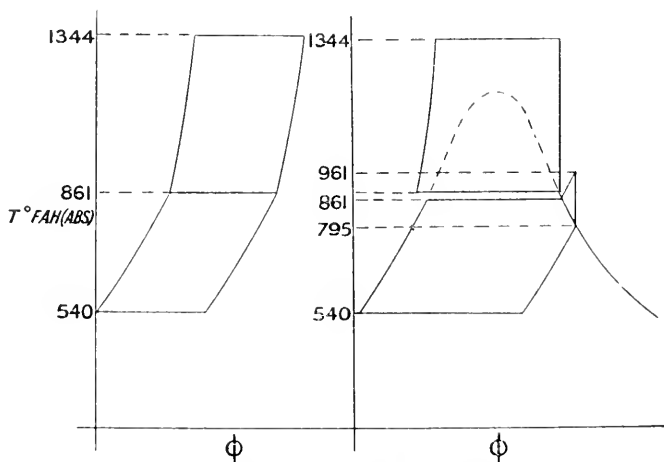


FIG. 27.—Turbine Cycles for Mercury and Steam.

For the purposes of comparison, the mercury cycle is plotted for that mass of mercury which could be vaporized by the heat in unit mass of the furnace gases

above 884 deg. Fahr. As indicated above, the diagram for the steam relates to that mass which could be vaporized by the heat rejected by the mercury turbine. With such an arrangement, however, unless an inordinate amount of preheating were resorted to, the gases would escape at high temperature. Not only so, but since the liquid heat of mercury bears a comparatively small ratio to the latent heat of the vapour, there is little advantage in applying the principle of regeneration.

It therefore appears at present as though a satisfactory adjustment would consist in using the flue gases to heat up the water from that temperature at which the steam just became dry and saturated after expansion from a moderate degree of superheat, for superheating the steam and for heating up and vaporizing the mercury, regenerative heating being employed for the feed, as shown in the figure. Whatever heat might thereafter be available in the flue gases could be applied to preheating the air for combustion, and thus increasing the efficiency of the boiler.

The cycle would then take the form indicated in the diagram, the following calculations indicating the method followed.

Specific Heat of Furnace Gases.—Let the specific heat at constant pressure (C_p) of the furnace gases be represented by the equation

$$C_p = (0.2309 + 0.0000159 T).$$

Due to the fact that actual gases are not ideal in the sense in which this was understood in connection with the molecular theory, a certain proportion of the communicable energy of the molecules takes the form of internal vibrations of the atoms of which the molecule is composed. Thus if we can conceive of the molecule of a gas as being composed of a group of atoms held together by quasi-elastic restraints, which permit a more or less rapid relative displacement of a vibratory character on the part of the atom with reference to the group (a motion which becomes of greater importance as the temperature rises), we have an explanation of the presence

of the second term in the above equation. At very high temperatures it becomes desirable to add a third term involving a function of the square of the absolute temperature.

Ideal Efficiency of Turbine Cycles for Mercury and Steam.—Our first concern, therefore, is to find that temperature at which the steam just becomes dry and saturated, after expansion from a pressure of 250 pounds per square inch and, let us say, 100 degrees of superheat. This may very conveniently be done by reference to the Mollier chart, the pressure at the point in question being found to be 110 pounds per square inch to the nearest 5 pounds per square inch. This corresponds to a temperature of 335 deg. Fahr. The heat ideally available in the hot gases is therefore

$$\int_{T_o}^{T_1} T d\phi = \int_{T_o}^{T_r} T \frac{C_p dT}{T} = \int_{T_o}^{T_x} (A + BT) dT.$$

Inserting numerical values we have

$$\int_{795}^{3460} (0.2309 + 0.0000159T) dT,$$

which gives

$$0.2309 \times 2665 \times \frac{0.0000159(3460^2 - 795^2)}{2} \\ = 705 \text{ B.Th.U.}$$

The next step is to find the heat rejected by the mercury turbine per pound of mercury passing through the cycle.

From the Mollier chart of entropy and total heat for mercury vapour, the dryness is found to be 0.70. Hence the heat rejected is $0.70 \times 128.2 = 89.7$ B.Th.U. This, by assumption, is wholly employed in the vaporization of the steam, and thus the mass of steam concerned per unit mass of mercury is the ratio of the heat given up by the mercury turbine to the heat of vaporization of steam at 250 pounds per square inch, or $89.7/826.3 = 0.1086$ pound.

Thus we obtain the following equation :

Heat from furnace gases

$$= m_m(I_m + 0.1086 \times \text{heat to steam boiler}),$$

where I_m is the total heat of mercury vapour above the temperature of condensation, and m_m is the mass of mercury involved per unit mass of the furnace gases.

Thus we have

$$705 = m_m \{135.5 + 0.1086 (69.7 + 61.5)\}$$

$$\therefore m_m = \frac{705}{149.8} = 4.71 \text{ pounds of mercury}$$

and $0.1086 \times 4.71 = 0.512$ pound of steam.

The diagrams for both steam and mercury may now be plotted. The efficiency may also be determined as follows :

$$\text{Heat to mercury} = 4.71 \times 135.5 = 637 \text{ B.Th.U.}$$

$$\text{Heat rejected} = 0.512 \times 826.3 = 423 \text{ B.Th.U.}$$

$$\text{Heat used} = 214 \text{ B.Th.U.}$$

Similarly, heat to steam

$$= 0.512 (69.7 + 826.3 + 61.5) = 490 \text{ B.Th.U.}$$

Heat rejected from steam

$$= (\phi_{sup} - \phi_{r_{110}}) T_{540} \times 0.512 = 307.5 \text{ B.Th.U.}$$

Heat used = 182.5 B.Th.U.

Total heat turned into work is therefore $214 + 182.5 = 396.5$ B.Th.U. per unit mass of the furnace gases.

The overall efficiency of the binary vapour prime-mover is

$$\frac{396.5}{705} = 56.3 \text{ per cent.}$$

If the waste gases be used for preheating the air up to the temperature corresponding to a saturation pressure of 110 pounds per square inch, in the limiting case the efficiency of the boiler would be unity, since all of the heat supplied might then be transferred to the working substance.

In the foregoing no attempt has been made to estimate probable actual efficiencies, and the chief value of these analyses lies in their use for purposes of comparison.

Other factors remaining sensibly constant, the important gain to be anticipated from the use of a mercury turbine in association with a steam prime-mover is evident.

Description of the Apparatus.—As was to be expected, that design of boiler which functioned successfully for steam would not answer in the case of mercury. In the first place, considerations of cost render it advisable to reduce the mass of mercury in the system to a minimum. The temperatures reached in the mercury boiler are much higher than those for steam, and the chances of leakage or blowing out of a tube have to be reduced to a minimum, both on account of the cost and poisonous nature of the working substance. Further, the conditions under which mercury boils differ somewhat from steam.

While mercury is a good conductor of heat, it does not wet steel, and greater difficulty is experienced in causing heat to flow from the hot furnace gases to the mercury than to the water in a steam boiler. When mercury vaporizes, these conditions become rapidly worse, and hence it is advisable to expose those portions of the boiler containing liquid mercury to the greatest heat. This is accomplished in the later types of boiler proposed for this purpose, by means of a group of dead-ended tubes hanging in the combustion chamber, and having suitable internal arrangements for promoting the rapid circulation of the mercury. Since the specific gravity of mercury is relatively high, a large difference of head can be maintained between the tops and bottoms of the tubes, and hence boiling prevented until the upward current approaches the free surface.

The arrangement permits the free expansion of the tubes—a most important matter—in order to avoid any repetition of stress at the joints, tending to cause fracture. The joints of all pipes and containers carrying mercury are welded to ensure tightness.

Oil is used in the furnace for fuel. The hot gases pass upwards through the mercury boiler where evaporation takes place, the liquid mercury being heated up in a separate heater through which the gases are passed next

in order. The residual heat is then used to superheat the steam and to heat the feed-water, the gases passing off finally at a temperature of from 400 to 450 deg. Fahr.

The mercury vapour fed to the turbine is discharged directly from the rotor in the mercury condenser space against the boiler tubes, which, as in the case of the mercury boiler, hang downwards freely and are dead-ended. In the more recent development of the mercury turbine, the rotor is of the three-stage impulse type, and is mounted on the overhung end of the generator shaft.

With regard to the durability of mercury it appears to suffer little from oxidation at high temperatures, but it is considered advisable to reduce the amount of loss from this cause by introducing illuminating gas into all spaces containing mercury when the plant is shut down. Hydrogen and carbon monoxide are reducing agents for mercury and tend to prevent oxidation which, even in small degree, appears to render the metal adhesive and make it difficult to separate finely divided matter. The poisonous nature of mercury renders special precautions in its use highly desirable. In the existing plant at Dutch Point of the Hartford (Connecticut, U.S.) Electric Light Company no important trouble has arisen from this cause, but it is proposed to enclose all parts which carry mercury under pressure, and to subject the space so confined to stack suction, in those cases where mercury would not naturally be drawn away with the flue gases in the event of leakage.

Recent indications point to the early development of the process on a manufacturing basis.

THE STEAM-ENGINE

THE earliest attempts to convert the energy of steam into mechanical form were not, strange to say, along the lines of the simple steam-engine. The engine of Hero of Alexandria, who lived more than 2000 years ago, was a simple form of reaction turbine, and consisted of a spherical vessel mounted between centres and caused to rotate by the reaction of two jets of steam issuing from bent pipes at opposite ends of a diameter at right angles to the axis of rotation. The world had to wait 1800 years before the steam prime-mover played an important part in industrial development. In the hands of Savery, Newcomen, and James Watt it assumed the essential characteristics of the steam-engine of our own time, the main line of advance being in the direction of greater economy, larger size, and refinement of mechanical detail.

Comparison of the Actual with the Ideal Cycle.—

In the ideal cycles for steam, already considered, no part of the heat energy of the working substance was supposed to be interchanged between the steam and the enclosing envelope, consisting of cylinder walls, piston face, etc. Further, it was imagined no irreversible processes of the nature of throttling actions took place. This, however, in practice, is far from the truth. Having, therefore, as our objective that design which most nearly fulfils the requirements of a perfect engine, let the records of actual engines be examined with a view to discovering in what directions improvement may be expected.

The most convenient method of analysing the action of the steam-engine is by means of the indicator diagram

in conjunction with some form of water meter for weighing or measuring the condensate or boiler feed, and a revolution counter for determining the speed. By these devices a record of the relation between the pressure and volume of the working substance throughout the stroke, and of the weight of steam used per stroke, is obtained.

A typical indicator diagram is shown in the figure. In order to locate the axis of zero pressure, the atmospheric line is drawn on the card before its removal from the instrument.

When the constant of the spring is known, or has

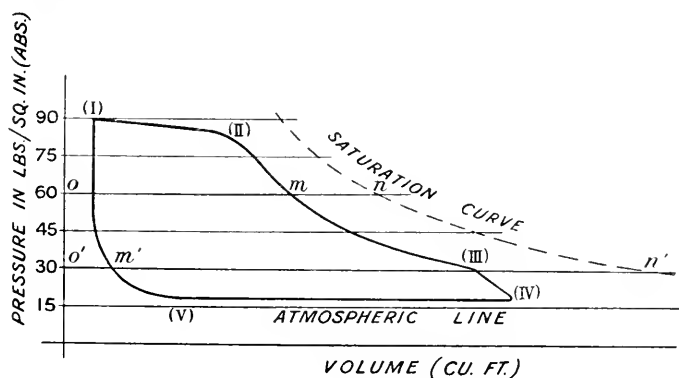


FIG. 28.—Steam-Engine Indicator Diagram.

been determined by calibration, the pressure scale may be laid down.

The axis of zero volume is fixed by reference to the length of the indicator diagram, between perpendiculars and to the known value of the clearance, usually expressed as percentage of the stroke.

With regard to the mass of the working substance, while it is obviously an easy matter to determine the average weight of the cylinder feed, the weight of the steam trapped in the cylinder has to be calculated by making assumptions which cannot be tested experimentally. It is customary to consider the steam present in the cylinder at the commencement of compression to be dry and saturated. Thus the weight of the cushion

steam, as it is called, and hence the total mass of the working substance taking part in the action may be estimated.

The next step is to plot a 'saturation curve'—that is, a curve exhibiting the relation between the pressure and volume of the total mass of steam involved, were it dry and saturated.

In order to simplify matters, the whole process may be thought of as taking place in the cylinder, heat being

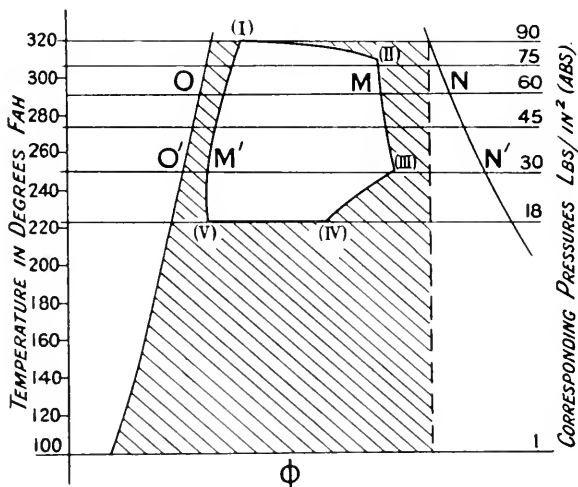


FIG. 29.—Temperature-Entropy Diagram for Actual Cycle of Steam-Engine.

added and rejected by arrangements similar to those imagined in the case of the Carnot engine.

The temperature-entropy diagram may now be plotted, from the consideration that for any point on the indicator diagram the dryness of the steam is measured by the ratios

$$\frac{OM}{ON}, \frac{O'M'}{O'N'}, \text{etc.},$$

whence the entropy is found by dividing ON , $O'N'$, etc. in the same ratio.

If now the figure so obtained be examined by reference to the ideal Rankine cycle, several points of

divergence causing important losses of work may be observed.

During admission the curve (i) (ii) slopes downwards, due to pressure drop as a result of frictional losses in the inlet channel. This is in the nature of a throttling action, and while, as we have seen, the total heat is carried over, the action is thermodynamically wasteful, since heat is removed from the steam in order to perform work against these resistances and is later restored at a lower temperature. Next we note that the curve (i) (ii) stops far short of the point reached by the corresponding curve on the ideal cycle. This loss is caused by initial condensation of the live steam against the comparatively cool and moist cylinder walls and piston face, and is perhaps the most serious of all the losses with which the ordinary steam-engine has to contend. While the curve of expansion (ii) (iii) invades the lost area to some extent, due to partial restoration of heat and consequent re-evaporation, the action is such as to cause loss of availability of the heat for the performance of work. Proceeding, the effect of incomplete expansion is seen to be responsible for the withdrawal of the heat equivalent of the area under (iii) (iv) down to abs. zero, of which a portion is still available for the performance of work. The temperature-entropy diagram further shows very clearly the ill-effects of high back pressure. It may be observed that as the pressure falls, the width of the corresponding temperature intervals increases. Hence a back pressure of the order of 18 pounds per square inch causes a large loss of work when compared with the ideal figure drawn, say, for 1.0 pound per square inch.

Development of the Steam-Engine.—From the time of James Watt these guiding considerations, notably the ill-effects of cylinder condensation, have been recognized and have been the determining factors in the evolution of the steam-engine. He it was who contributed the separate condenser and the steam-jacket, both devices aiming directly at a reduction of the condensation losses. It might appear at first sight as though the addition of the steam-jacket merely increased the area exposed to

radiation and convection. Its effect, however, is, on the whole, usually beneficial, since it tends to keep the cylinder walls dry. When in this condition the tendency towards initial condensation is greatly reduced.

Superheating has a similar effect, and represents at the same time a slight thermodynamic gain, in that a part of the heat taken in, in this way, has a greater availability for conversion into heat. While a moderate degree of superheat may not entirely prevent initial condensation, there is a strong drying tendency, and heat lost on entry is restored at an earlier stage of the expansion. Compounding possesses both mechanical and thermodynamic advantages. Each cylinder is subjected to but a fraction of the total temperature range of the steam, and the effect of two, three, or four cranks, as the case may be, is to even the torque and reduce the size and weight of the working parts. Even with these arrangements, however, large economies are still possible.

The Uniflow Engine.—Perhaps the most notable advance in steam-engine design in recent years is the uniflow engine.

The important feature which differentiates this class of engine from its predecessors is the use of a central exhaust belt in a comparatively long cylinder, fitted with a trunk piston, which functions also as a release valve. Steam enters at both ends of the cylinder alternately, by short passages, and flows towards the central ring of exhaust ports, which are uncovered as the trunk piston nears the end of its stroke.

A high vacuum in the condenser is essential unless auxiliary exhaust valves be employed, since, on the return of the piston, compression begins very early in the stroke and a back pressure in excess of the inlet pressure would result with clearances of normal proportions.

A characteristic diagram from an engine of this type is shown in the fig. 30. A consideration of this indicator card shows that the temperature range of any circumferential belt of the cylinder surface is smaller than in the case of the ordinary steam-engine. Under steady running conditions a temperature gradient

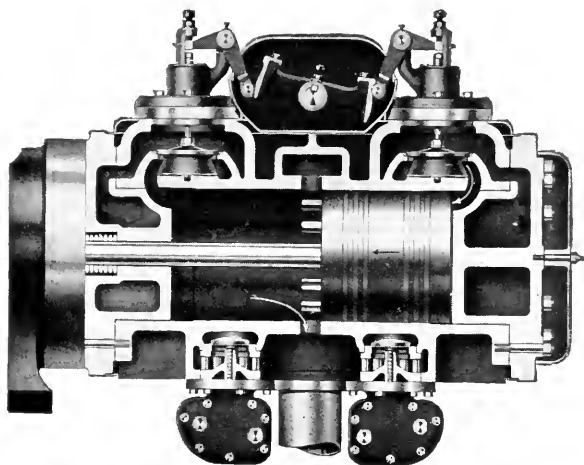


FIG. 31. Admission on dead centre at back end ;
exhausting from crank end.

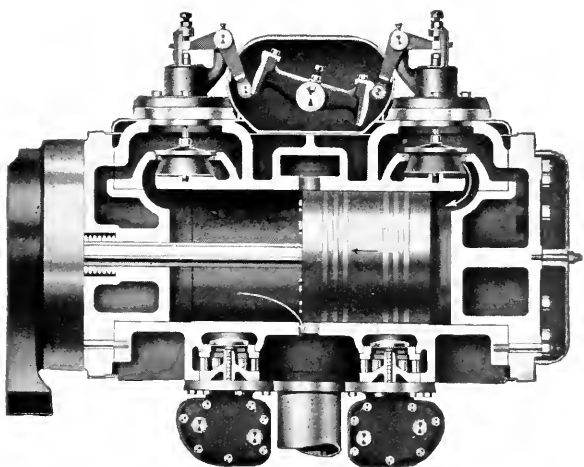


FIG. 32. Admission valve, back end, fully open ;
compression, crank end, about to commence.

(By courtesy of Skinner Engine Company Erie, Pa.)

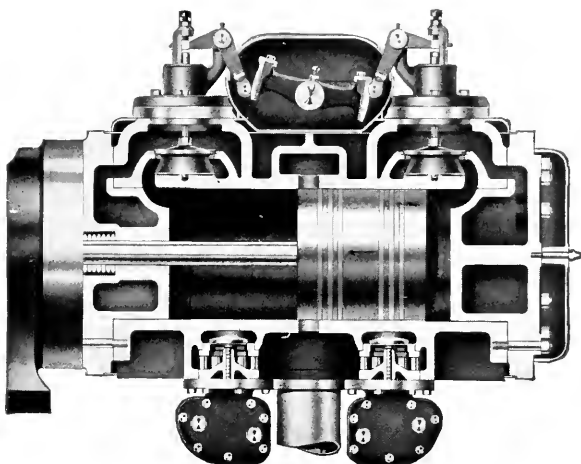


FIG. 33. Cut-off at back end ; compression in crank end.

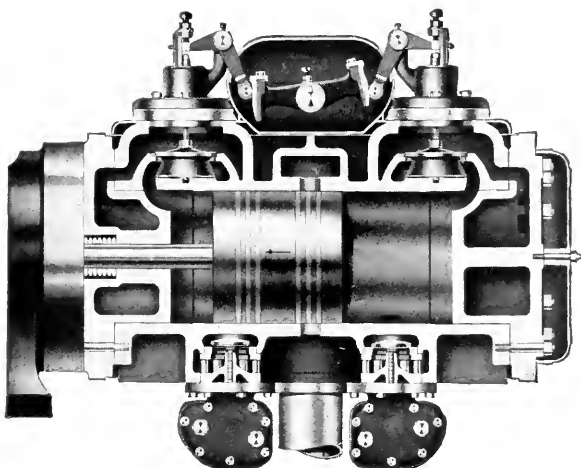


FIG. 34. Expansion in back end ; compression in crank end.

(By courtesy of Skinner Engine Company, Erie, Pa.)

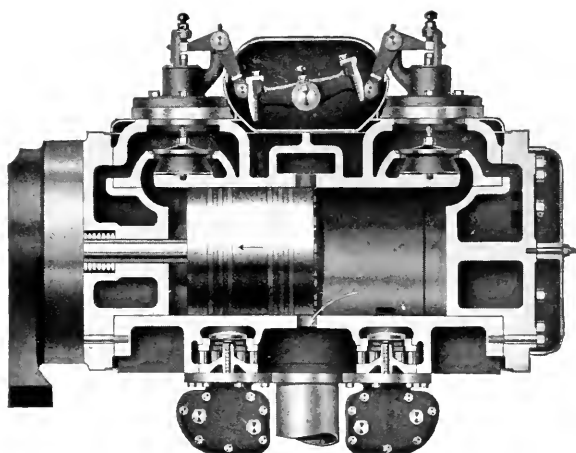


FIG. 35. Release in back end ; compression in crank end.
(By courtesy of Skinner Engine Company, Erie, Pa.)

PLATE III.

from the inlet ends to the central exhaust ports is established.

Thermodynamically, the arrangement is highly satisfactory, and while at a disadvantage in some respects mechanically, the saving in steam is remarkable. The best results are obtained with high vacuum, but some of the advantages are retained in non-condensing service, auxiliary valves being employed to obviate the necessity for large clearance spaces.

Figs. 31-35 illustrate the 'Universal UnafLOW' engine, manufactured by the Skinner Engine Company of Erie,

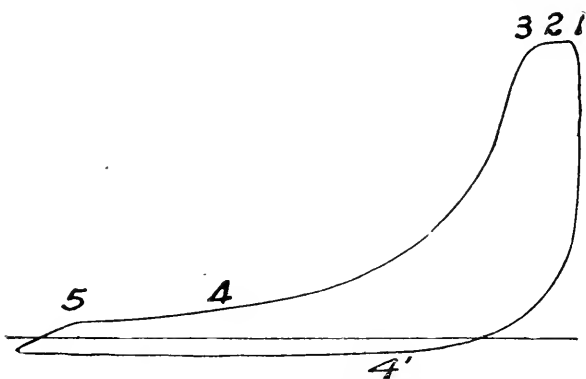


FIG. 30.—Indicator Diagram of 'Universal UnafLOW' Engine.

Pennsylvania, U.S.A., the relations of the various parts of the mechanism at the several events of the cycle being well seen.

Temperature-Entropy Diagram for Uniflow Engine.—Fig. 36 shows the characteristic form of temperature-entropy diagram for an engine in which the initial losses due to cylinder condensation have been reduced to a minimum. In order to interpret the figure, it must be remembered that the working substance is divided between the cylinder and the boiler on the one hand, and between the cylinder and the condenser on the other. Consider the curve (i) (ii). During this stage the mass of the working substance in the cylinder is increasing. From (ii) to (iii) the total mass of steam taking part in the action is present in the cylinder. From

(iii) to (v) the mass is again diminishing, and from (v) to (i) we have only the cushion steam in the cylinder. If it be desired, for example, to determine the nature of the latter action in which the clearance steam alone is concerned, it is necessary to plot an adiabatic for comparison. This may be done by setting off from the water curve for the cylinder feed alone, another curve such that the horizontal displacement is constant and equal to the entropy difference between the cylinder feed and the total

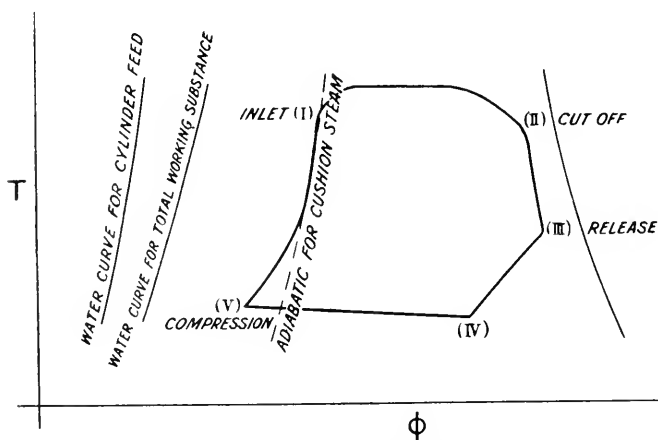


FIG. 36.—Temperature-Entropy Diagram from Indicator Card of 'Universal Unaflo' Engine.

mass taking part, at the commencement of compression. If the diagram has been plotted per unit mass, then the water curve for the cylinder feed is plotted for that mass which bears the same ratio to unit mass that the mass of the cylinder feed bears to the total mass of the working substance. In general, the state point on the temperature-entropy diagram corresponding to any given co-ordinates of pressure and volume gives a measure of the sum of the entropies of the parts into which the working substance is divided. If the condition of one part be known, then that of the other may be found.

The Performance of Steam-Engines.—The usual commercial method of stating engine performance is by means of water rate—that is, the number of pounds of

steam consumed per indicated or per brake horse-power hour. In order to make comparisons it is necessary to apply suitable corrections to take account of varying initial total heat of the steam. By stating the performance as an efficiency, or better still as a relative efficiency, the reference being the Rankine standard, any ambiguity is avoided.

The observations which must be made in an engine test consist of the mass and condition as to dryness or superheat of the steam supply, the indicated horse-power and the brake horse-power.

Further, if a complete heat account be desired, it is necessary to measure the heat given up to the condenser and to the steam jackets, loss in hot well, etc., the balance of heat otherwise unaccounted for being put down to radiation, leakage, etc.

In determining the dryness of the feed, the steam sample is commonly throttled in a suitable calorimeter, a slight amount of superheat being produced in the process. As the condition of the steam is readily measured after throttling by means of a thermometer and pressure gauge, the total heat before throttling is thus found, and the dryness calculated. Some care is necessary in securing a fair sample of steam from a pipe. Even so the results are commonly unreliable. The mass of steam supplied in any given time is easily found by weighing or gauging the condensate or by measuring the boiler feed. When the latter plan is adopted, the test must, in general, be of greater length, in order to render any error in estimating the level of the water in the boiler inappreciable. The indicated horse-power is frequently used as an index of engine performance on account of the ease with which the test can be carried out.

The essential features of a satisfactory indicator are a carefully fitted cylinder and piston, together with a linkage which is free from backlash and has the smallest possible inertia. The spring should be removed as far as possible from the effects of temperature changes, and calibration should be carried out at the temperature which the spring assumes in service. An inextensible cord should be

employed to drive the drum, as quite important errors are liable to arise from the use of a long cord possessing an appreciable amount of elasticity. Care should be taken in use to see that the control cock is fully opened and throttling thereby avoided.

The brake horse-power of engines of moderate power is usually measured by some form of band or water brake, which gives the torque. Electro-magnetic devices may be used in the same way, the prime-mover driving the armature, while the field is prevented from rotating by suitable balance weights. The torque being known, the power output is found when the number of revolutions per minute is known; thus

$$\text{Horse-power} = 2\pi FN \div 33000,$$

F being in pounds-feet, and N in revolutions per minute. For large powers absorption dynamometers are unsuitable, and some form of transmission dynamometer must be employed. For instance, in estimating the net power delivered to the propeller of a ship, the twist of the shaft over a short length is measured by noting the angular deflection of a beam of light reflected from a mirror mounted on a collar on the shaft in such a manner that the relative angular displacement of two sections of the shaft a known distance apart causes rotation of the mirror about its axis. The twist of the shaft being proportional to the applied torque, the horse-power can be determined when the number of revolutions are known, and also the relation between the displacement of the beam of light (which flashes momentarily on the scale) and the torque. The latter constant is usually determined by a static test in the shops before assembly of the particular section of the propeller shaft upon which the torsion meter is mounted.

Temperature-Entropy Analysis of Indicator Diagram from ' Universal Unaflow ' Engine

By MESSRS. SKINNER, Erie, Penna., U.S.A.

THE following data were obtained from a test of a 21-inch bore, 24-inch stroke ' Universal Unaflow ' engine. The engine was direct-coupled to a 200-k.w. General Electric Company's generator.

Hourly water rate, less jacket steam	4371 pounds
Revolutions per minute	197
Percentage moisture in steam as supplied	1.8

From the constants of the engine and the known speed, we find that the average feed per stroke is

$$\frac{4371}{23640} = 0.185 \text{ pound.}$$

The volume of the cylinder, neglecting for this purpose the volume of the piston rod, is

$$\frac{3.14 \times 1.75^2}{4} \times 2 = 4.81 \text{ cubic feet.}$$

The total clearance is given as 6.68 per cent. of the stroke. Hence the total volume of cylinder + clearance is

$$1.0668 \times 4.81 = 5.13 \text{ cubic feet.}$$

In order to determine the total mass of the working substance, some assumption must be made with regard to the condition of the clearance steam. For the purposes of this analysis let it be supposed to be dry at the end of compression. From the indicator card provided, this latter point would appear to occur at, or near, 118 pounds per square inch. The volume of unit mass of dry and saturated steam at this pressure is 3.79 cubic feet.

Hence the mass of the steam shut in the clearance becomes

$$\frac{V_{cl}}{V_{118}} = \frac{0.321}{3.79} = 0.0847 \text{ pound.}$$

Thus the total mass of the working substance is 0.27 pound.

Let, now, the indicator card be taken and scales of pressure and volume set up. The card should be mounted on the drawing board with the atmospheric line horizontal. With the set square, two vertical lines are then drawn to touch the diagram at either end. The horizontal distance

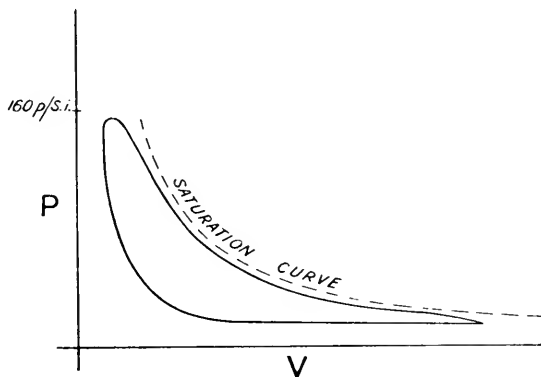


FIG. 37.—Indicator Card from 'Universal Unaflo' Engine under test.

between these lines is then a measure of the volume of the cylinder.

Taking 6.68 per cent. of this distance, erect the line of zero volume.

The axis of zero pressure is then drawn, the indicator spring constant being known. Using an 80 pounds per square inch per inch spring, calibration showed that a correction factor of 1.03 should be applied. This means that for every inch vertically the pressure changes by 80 times 1.03, equals 82.4. Diagonal lines will be found very convenient for setting off both the pressure and volume scales.

The saturation curve is then plotted, and the $T\phi$ diagram drawn as above explained.

THE STEAM-TURBINE

THE failure of the steam-engine to make complete use of the heat available in the steam, as shown clearly on the temperature-entropy diagram, gave the steam-turbine its opportunity. At one time used to complete the expansion of the steam from the exhaust of ordinary reciprocating engines for marine purposes, its sphere of usefulness has grown to include the whole range of expansion. Not only are the losses peculiar to the steam-engine largely avoided, but a much larger output of power per cubic foot of space occupied is obtained. The essential distinction between the steam-turbine and the steam-engine is that, whereas in the latter the work available is transferred directly to the mechanism, in the former the available energy of the working substance is employed to set the steam itself in motion. The discussion therefore naturally falls into two parts—namely, a consideration of the means whereby the maximum amount of heat ideally available in the steam may be transformed into kinetic energy of the moving jet, and the ways in which that energy may be recovered in the form of mechanical work.

Turbine Nozzles.—Referring to the formulæ already developed relating to the flow of fluids, we note that if the ratio in which the pressure falls is less than that given by the expression

$$D = \left(\frac{2}{n + 1} \right)^{\frac{n}{n-1}}$$

the nozzle will have a throat and will consist of both a converging and a diverging portion. In practice, it is found sufficient to provide a smooth, well-rounded entrance

which is short compared with the diverging portion—generally a simple conical frustum.

Fig. 38 shows the profile of the nozzles used on the De Laval turbine manufactured by Messrs. Greenwood and Batley, Leeds.

This is an impulse type of machine—that is, the available energy is developed wholly in the nozzle, the pressure of the fluid in which the rotor revolves being sensibly constant. In the reaction type of turbine, the channels formed by the blades of the rotor also function as nozzles, and a drop in pressure and increase in the velocity of the steam takes place in them. In order to calculate the discharge from a nozzle the formula already found may be applied—namely, mass of discharge in pounds per second

$$= M_{dis} = A_t \left\{ \frac{P_1}{V_1} \frac{2gn}{n-1} \right\}^{\frac{1}{2}} \left\{ D^{\frac{2}{n}} - D^{\frac{n+1}{n}} \right\}^{\frac{1}{2}}.$$

Since supersaturation persists in general at least as far as the throat, the appropriate value of n will be 1.3. The formula then becomes, on reduction—

$$M_{dis} = 0.3155 A_t \sqrt{\frac{p_1}{V_1}},$$

p being in pounds per square inch and A_t the area of the throat in square inches.

In order that the exit area of the nozzle may be suitably proportioned, the velocity and condition of the steam on leaving the nozzle must be known. In order to determine these quantities recourse may be had to the Mollier chart of $I\phi$, account being taken of the combined influence of supersaturation and friction. In practice it is impossible to separate these two effects, and observation alone can determine in any given case what proportion of the heat available is returned to the steam. This being known, the result may be applied to other similar cases and the velocity and dryness on exit calculated. Referring to fig. 24, (ii) (iv) is the ideal heat drop under stable conditions, (ii) (iii) is that for supersaturated steam, down to the pressure at the throat.

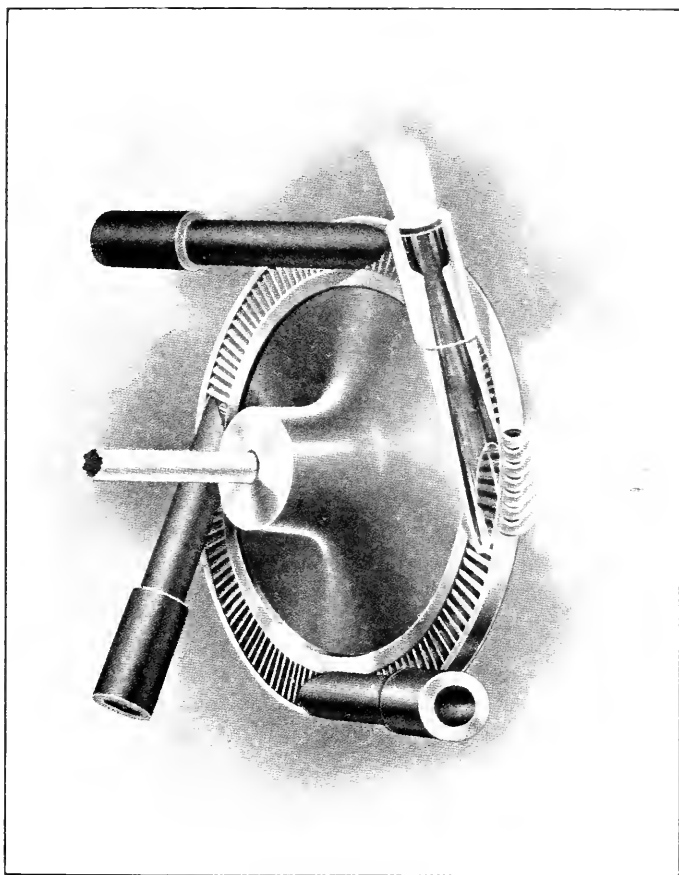


FIG. 38. Expansion Nozzles and Rotor of Single Stage Impulse Steam Turbine, by Messrs. Greenwood and Batley, Leeds.

PLATE IV.

If reversion be supposed to occur at that section, the state point moves to (iii)'. (iii)' (v) would give the heat drop from the throat to the exit, friction being absent. Due to the restoration of a certain amount of heat to the steam the state point departs to the right, the steam becoming drier than for isentropic expansion and the heat available for the performance of work diminishing. The velocity is found from

$$V_{ex} = \sqrt{2gJ(I_{(ii)} - I_{(v)'})}$$

The total mass of the discharge from the nozzle having been calculated, and the dryness and velocity of the steam at exit being known, the area at exit is given by

$$A_{ex} = \frac{V_{ex} M_{dis}}{v_{ex}}$$

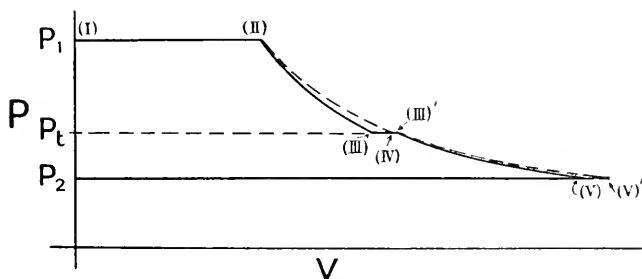


FIG. 39.—The Relations of the Pressure and Volume of Steam during expansion in a nozzle.

The matter may also be studied by reference to the PV diagram.

In the early stages of expansion, the steam follows a law of the form $PV^{1.3} = \text{a constant}$, or, more precisely, $P(V - V_w)^{1.3} = \text{constant}$, V_w being the volume of water at freezing-point. This part of the expansion curve is represented by (ii) (iii) on the diagram.

On reversion the point (iii) moves to (iii)', the work lost by supersaturation—namely, the area (ii) (iii) (iv)—appearing as heat in the steam. Thereafter the effect of friction, which has been disregarded in the expansion to the throat, is to increase the apparent work, but to diminish the net work done upon the steam, since the

area (iii)' (v) (v)' is obtained at the expense of the kinetic energy of the steam. Actually no such complete reversion to equilibrium conditions occurs as suddenly as suggested by the figure. Partial reversion probably occurs before the passage of the throat, and a certain amount of supersaturation doubtless persists as far as the later stages of expansion. The experiments of Wilson indicate that in the absence of suitable nuclei, such as are ordinarily present in the form of particles of dust, condensation does not appear to begin until the ratio of the actual pressure to that of the pressure corresponding to the temperature reached during the expansion is eight. This is known as the Wilson limit.

An example may serve to make this clear.

Suppose it is desired to determine the pressure at which reversion will occur in the case of steam initially dry and saturated at a pressure of 100 pounds per square inch.

For the expansion of the supersaturated steam, we have a law of the form $PV^{1.3} = \text{constant}$.

Also, on the assumption that supersaturated steam follows the law relating the pressure volume and temperature of a gas, we have $PV = RT$.

Now one-eighth of 100 pounds per square inch is 12.5. The temperature corresponding to this pressure is 204 deg. Fahr., nearly.

From the two laws above cited we obtain

$$\left(\frac{P_r}{P_1}\right)^{\frac{3}{13}} = \frac{T_r}{T_1},$$

P_r meaning the pressure on reversion, and
 T_r meaning the temperature on reversion.

$$\begin{aligned}\therefore P_r &= P_1 \left(\frac{T_r}{T_1}\right)^{\frac{13}{3}} \\ &= 100 \left(\frac{664}{788}\right)^{4\frac{1}{3}} \\ &= 47.45 \text{ pounds per square inch.}\end{aligned}$$

This example shows that reversion occurs almost immediately after the passage of the throat. While this view is

not strictly in accord with the facts, for the purposes of calculation it may be assumed that in a convergent-divergent nozzle friction may be neglected as far as the throat, and stable conditions of thermal equilibrium, in which friction is the important factor causing departure from the isentropic, may be assumed to hold thereafter.

Friction in Turbine Nozzles.—Various factors affect the actual efficiency of the nozzle, causing, in general, an increase or diminution of the loss due to friction. The initial quality of the steam, the length and condition of the internal surfaces of the nozzle, and the magnitude of the heat drop, all affect the final result. A formula, due to Mr. H. M. Martin, may be used to find the efficiency :

$$\text{Effy} = \{102.7 - 0.06 (I_1 - I_2)\} \text{ per cent.}$$

$I_1 - I_2$ being the heat drop. This expression should be used only in those cases in which the heat drop is in excess of the drop to the throat. In general, in those cases in which a divergent portion is necessary for complete expansion an average efficiency of 90 per cent. may be anticipated, with a range of 5 per cent. on either side of the mean. Should the exit area of the nozzle be improperly proportioned, important losses are liable to occur. If, for instance, the exit area is too large, the steam pressure falls below that of the back pressure before discharge, and over-expansion results. This has more serious effects than under-expansion, of which a certain proportion may be tolerated, without important loss, at full load, so that at lighter loads the error tends to diminish, due to throttle valve control of the flow.

Turbine Blading.—Though not strictly a thermodynamical problem, the means whereby the kinetic energy of the moving column of steam is extracted may appropriately be considered here.

Let a jet of steam impinge upon the inner tip of a semicircular vane as shown in fig. 40.

Consider the vane in the first instance to be at rest. Then, if the effects of friction, etc., be neglected, the

steam passes from the vane with its original velocity undiminished in magnitude but reversed as to direction.

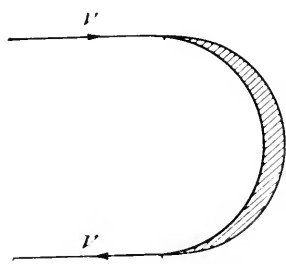


FIG. 40.—Flow of Steam across a Stationary Vane.

Let now the vane move with one-half the velocity of the steam, and consider what happens. The relative velocity of vane and steam on entry is one-half v . If, as before, friction be neglected, the relative velocity will remain constant in magnitude, although the direction of relative motion will change in the passage across the vane.

Referring to fig. 41, the actual velocity of the steam at any point is determined from the consideration that the vector sum of the velocity of the steam relative to the vane and the velocity of the vane, is the velocity of the steam relative to the frame of the machine.

If any imaginative difficulty is experienced in dealing with the problem of relative velocities, a useful rule is the following: Impress upon both bodies the velocity of the body of reference reversed. This brings the body of reference to rest without altering the relative velocity of the two bodies.

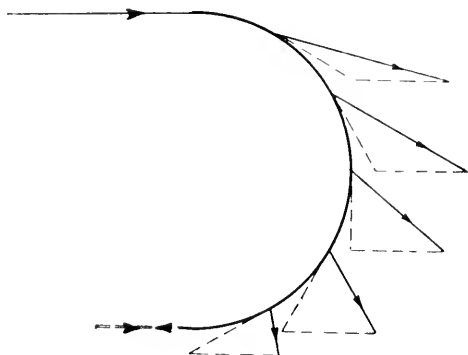


FIG. 41.—Flow of Steam over a Vane moving with half the velocity of the jet.

When, therefore, the steam has completed the circuit of the semicircular vane its actual velocity is zero, and ideally all the kinetic energy originally possessed has been given up in doing work upon the vane.

In most types of turbine the jet enters the wheel at an angle to the plane of rotation and may pass through one or more moving rings alternating with fixed guide blades. Further, as already noted, the channels between both fixed and moving blades may function as nozzles and thus serve to increase the velocity of flow at the expense of a part of the available total heat. This latter class of machine is known as a reaction turbine in contradistinction to the impulse type in which the transformation of heat energy into kinetic energy takes place wholly in stationary nozzles.

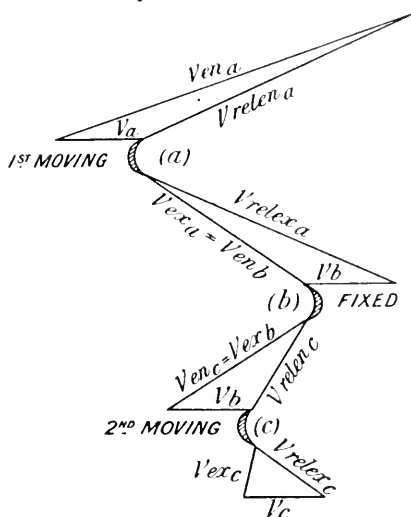


FIG. 42.—Shows Entry and Exit Velocity Triangles for two rows of moving and one row of fixed blades.

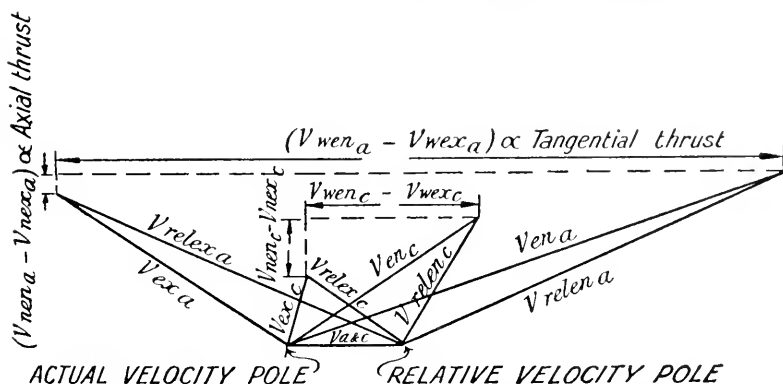


FIG. 43.—Convenient method of drawing Velocity Triangles for a series of blade rings and from which the work done upon each ring and the end thrust may be readily determined. The diagram is drawn for an axial flow machine.

Work done upon Turbine Blading.—Consider the equilibrium of any element of the jet of steam which

flows over the surface of a turbine blade. Let the change in velocity which it suffers in dt second be dv feet per second. Then the reaction of the segment of the vane with which it was in contact during the action is obtained from Newton's law,

$$P = \frac{ma}{g},$$

P being in pounds when m is in pounds. If in the present instance m pounds of steam are flowing per second, the mass of the element in question is $m dt$.

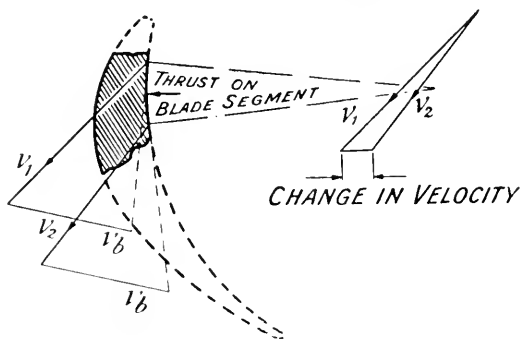


FIG. 44.—Determination of the Pressure upon an Element of a Turbine Blade.

Hence the pressure on the vane segment is

$$dP = \frac{m dt dv}{g dt}$$

which becomes

$$dP = \frac{m dv}{g}.$$

Now velocities are vector quantities and must be summed by the parallelogram law. When this is done, the total pressure upon the vane is given by

$$P = \frac{m}{g} v_{imp}$$

where v_{imp} is the total change of velocity of the steam.

The line of action of the resultant pressure on the vane

whose equilibrium is under consideration is parallel to the direction of the impressed velocity, and will, in general, be inclined at an angle to the plane of rotation of the wheel, in the case of axial flow machines. In order to find the point of application of this resultant, the vane

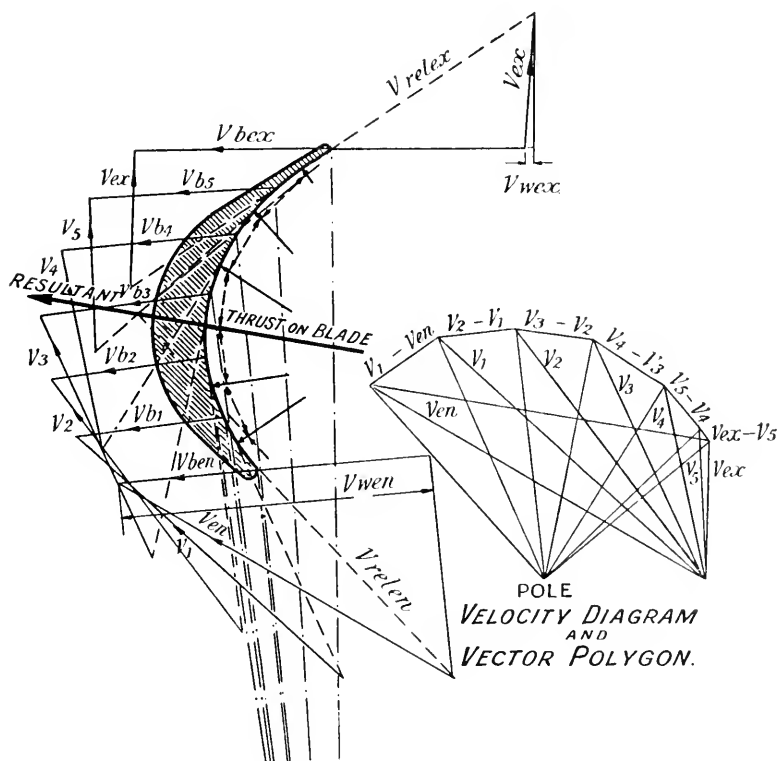


FIG. 45.—Illustrating the General Principles of the Flow of a Fluid over a Vane.

may be divided into a series of segments, and the force upon each determined. The resultant of this system then intersects the profile of the vane at a point such that the effect of the resultant acting alone is the same as would be observed by the several components acting separately. The stresses induced in the vane may then be computed by the principles of mechanics, the vane as a whole, of course, being considered.

In order to find the work done upon the blade, the component of the pressure between the vane and the steam in the direction of motion of the blade must be determined. The total change of velocity in the direction of motion of the vane is given by the algebraic difference of the tangential components of the actual velocities at entry and exit. These quantities are known respectively as the 'velocity of whirl at entry' and the 'velocity of whirl at exit.'

Hence we have the following equation, which is of considerable importance :

$$\text{Work done per pound of steam} = \frac{v_b}{g} \{v_{wen} - v_{wex}\},$$

in which v_b is the velocity of the blade,

v_{wen} is the velocity of whirl at entry,

and v_{wex} is the velocity of whirl at exit.

Regard must be paid to the sign of the velocity of whirl at exit. If the component of the actual velocity at exit in the direction of motion of the exit tip is in the opposite direction to v_{wen} , then the quantity within the brackets above is an arithmetical sum.

For a radial flow machine the repression becomes,

$$\frac{1}{g} \{v_{wen} v_{b_1} - v_{wen} v_{b_2}\}$$

in which v_{b_1} is the velocity of the tip of the blade at entrance,

and v_{b_2} is the velocity of the tip of the blade at exit.

The component of the pressure between the steam and the vane at right angles to the latter's motion does no work, but, in the case of axial flow machines, produces an end thrust upon the rotor, which must be taken up by suitably proportioning a dummy piston upon the rotor spindle or by a thrust block.

Friction in Turbine Blading.—Due to shock at entry and other causes, a part of the kinetic energy of

the steam flowing over the surface of a turbine blade is turned into heat. Hence the velocity at exit is always less than for frictionless flow. For impulse machines, in which the effect is not masked by increase in the velocity of flow due to drop in pressure across the ring, Professor Goudie gives

$$v_{relr} = \left\{ 0.75 + \frac{v_{relen}}{23,000} \right\} v_{relen}.$$

THE WESTINGHOUSE IMPULSE-REACTION TURBINE

THE following account may serve to give some idea of the design of a modern turbine.

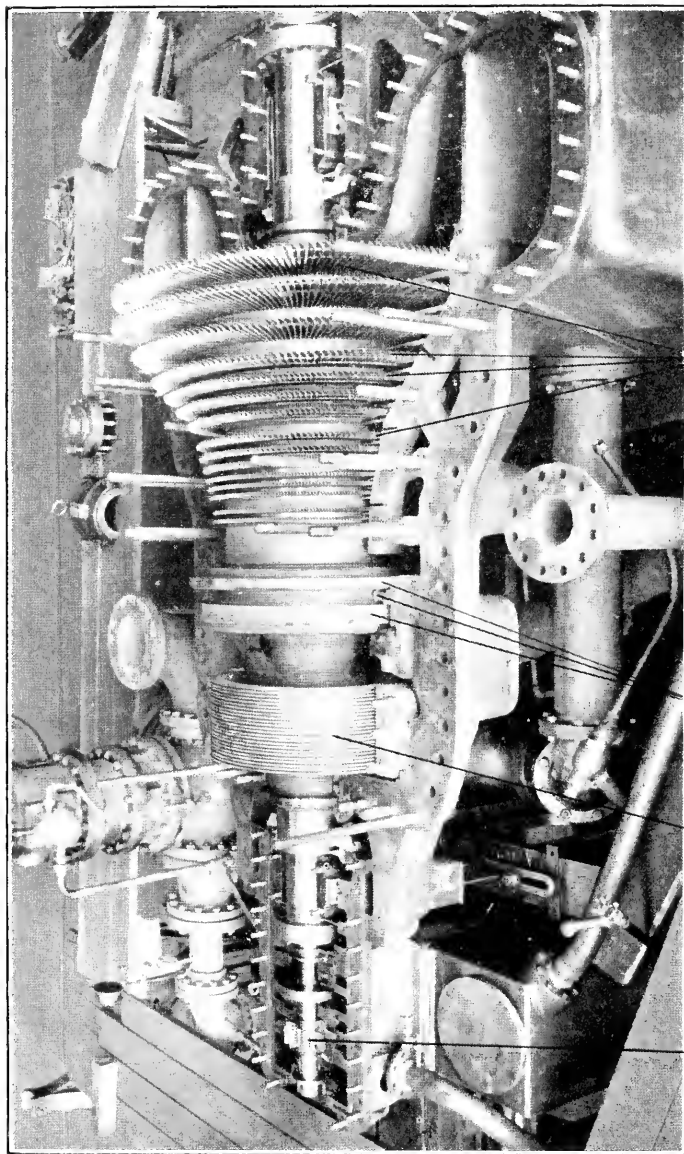
The Turbine Stator.—As may be seen from fig. 46 the turbine cylinder is split horizontally in two sections.

The top half of the cylinder is provided with eyebolts for lifting, and a sentinel valve which operates if the pressure should build up in the exhaust beyond a safe limit. The cylinder is conically bored, providing a uniformly expanding passage for the steam flow through the blading. This form of profile tends towards the reduction of eddy losses.

All the stationary blades, both impulse and reaction, are installed directly in the stator. The exhaust chamber side walls of the latter are of sufficient strength to take the entire weight of the condenser, which may thus be bolted directly to the exhaust flange, thus eliminating an expansion joint.

The machine has no bedplate. The bearing pads are cast integral with both the turbine and generator stators, and rest directly on the foundations. The governor end of the turbine rests upon a bearing pedestal bolted to the upper flange of an **I** beam, which is secured in the foundation cross-wise to the turbine axis. The longitudinal expansion of the turbine due to temperature changes is then taken care of by the deflection of the beam in the horizontal plane.

Four bleeder openings are provided in the turbine for the extraction of steam which may be used for feed-water heating.



IMPELLER OIL PUMP.

DUMMY PISTON
AND
LABYRINTH SEAL.

REACTION BLADING.

FIG. 46. Westinghouse Impulse Reaction Steam Turbine.

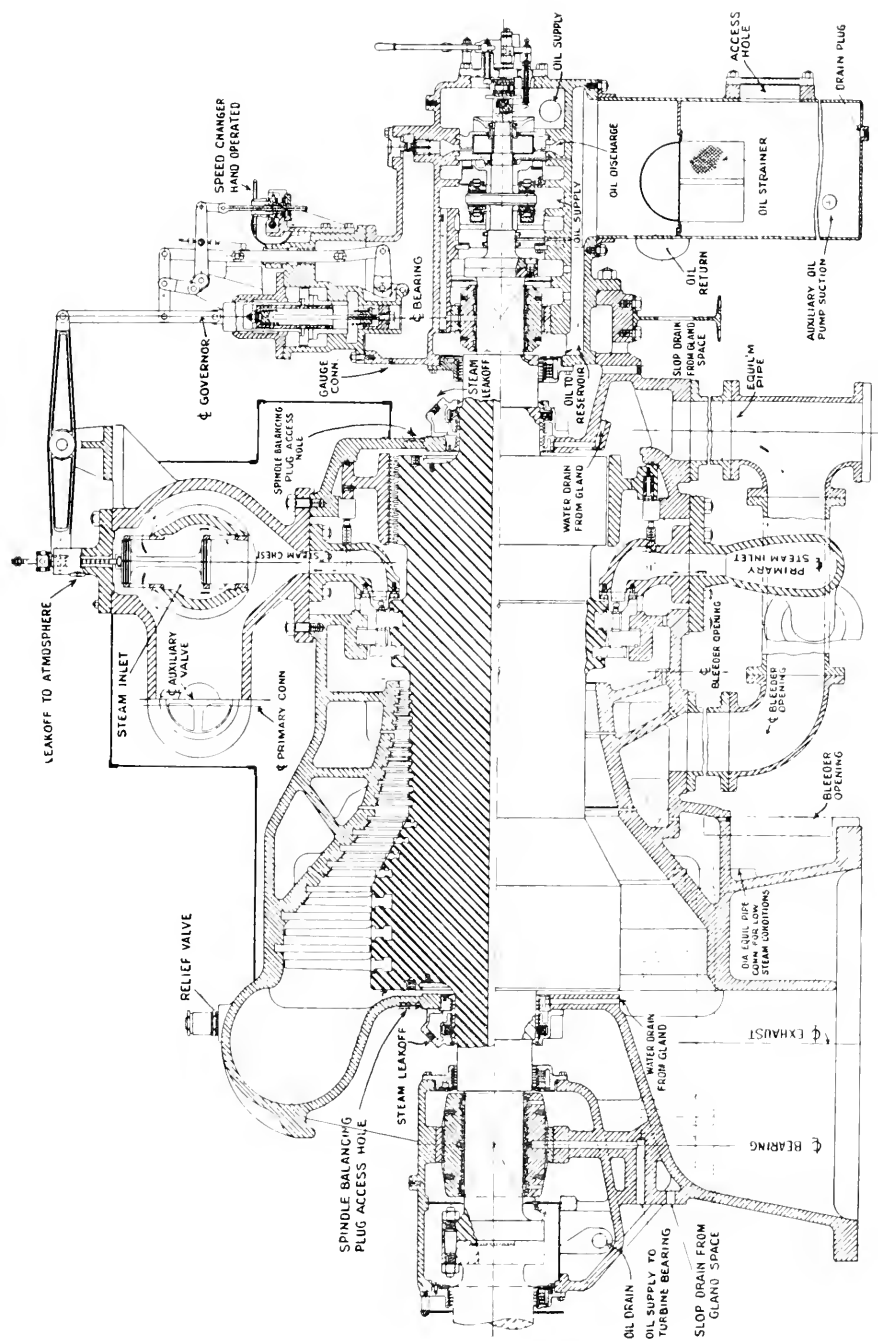


Fig. 47.—Cross-sectional Elevation of Westinghouse Impulse-Reaction Steam-Turbine.

These openings, it may be observed, are tapped off at progressively lower pressures, and consequently temperatures, and this device is in accord with fundamental thermodynamic principles of economical operation. If such openings could be indefinitely increased in number, the cycle would approximate more and more closely to the regenerative cycle of Stirling, and the efficiency, for the assigned temperature limits, would be that of the Carnot cycle.

The Turbine Rotor.—The rotor consists of two high-grade steel forgings, carefully annealed and accurately machined all over to ensure uniformity of centrifugal stresses and balance.

The two sections are forced together with a long spigot centring fit. (The joint is not shown in the cross-sectional elevation.) A sufficient number of heavy tap bolts are provided to hold the two parts securely at all times, without the aid of the press fit.

The governor end of the spindle is machined to form a dummy piston as shown in fig. 46. This piston is provided to counterbalance the end thrust, due to the difference in pressure of the steam entering and leaving the reaction blading, and to the axial dynamical thrust upon the blades. Any additional end thrust not counterbalanced by the piston is amply taken care of by the Kingsbury thrust bearing. This thrust block operates upon the same principle as the Mitchell, and depends upon the high resistance to pressure offered by wedge-shaped oil films, drawn in between a series of tilting bearing pads and the thrust collar.

The rotor carries two rows of impulse blades, followed by eighteen rows of reaction blades. The complete spindle is accurately balanced in a dynamic balancing machine before mounting.

In order to reduce leakage of steam past the dummy piston, the usual packing methods employed in reciprocating engine practice are not used. Instead, a series of deep collars are machined on the piston as shown in fig. 48, and these rotate within corresponding fins made of brass strips which are secured around the inner

circumference of the stator cylinder. The clearances between the stationary and rotating elements are made as small as practical considerations will permit. Any steam which escapes through the labyrinth is returned to the low-pressure end of the turbine.

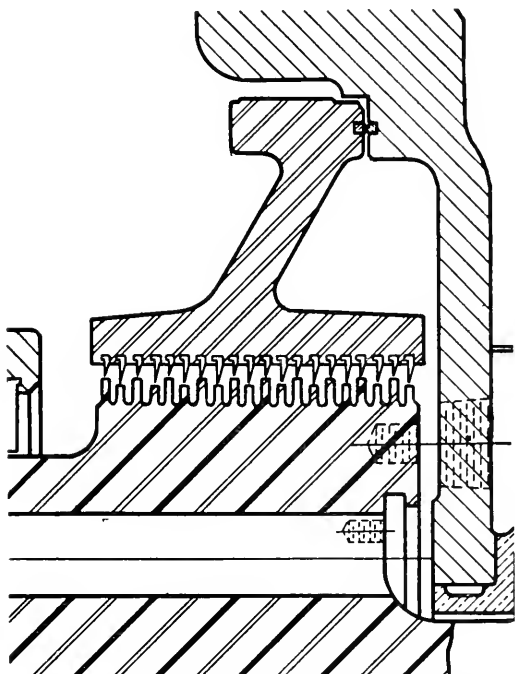


FIG. 48.—Peripheral Labyrinth Seal on Dummy Piston.

Steam Chest and Admission Valves.—The turbine is equipped with cast-steel steam chest superimposed on the cylinder cover. It contains two admission valves of the balanced poppet type, which are hydraulically operated by oil under pressure and under governor control. These valves control admission of steam to two sections of the nozzle chamber, around which are corresponding groups of nozzles in which the steam is partially expanded and directed upon the first row of impulse blades. With this arrangement of an impulse stage preceding the succeeding reaction stages, any

leakage over the tips of the blades in the early stages of expansion, when such leakage would be relatively difficult to keep within reasonable limits, is entirely avoided, since there is no pressure drop across the impulse rings.

The valves are made of steel and fitted into steel cages secured in the steam chest.

Movement of the valves is caused by oil pressure acting upon an operating piston connected to the valves by linkage.

Pressure upon either side of the piston is controlled by a relay valve operated by the governor.

Blading.—The impulse blades are drop-forged from 5 per cent. electric furnace nickel steel, and are finished smooth and true. The first row is $2\frac{1}{2}$ inches wide—an unusually broad design intended to minimise the deleterious effects of erosion. These impulse blades are held in the rotor by a double-tongue groove and further secured by wedges, this feature being designed to prevent the localizing of vibration at the narrow section of the blades. Vibration in the direction of rotation is prevented by lashing the blades together at their tips by a strip of steel fitted in a rectangular groove cut in the ends of the blades and caulked and welded. The blades are lashed in groups to allow for expansion.

All reaction blading is made from manganese copper, except the longer, low-pressure moving blades, which are drop-forged from 5 per cent. nickel steel.

For use with the blades secured in the double-tongue grooves in the rotor, there has been developed a blade-locking device, whereby the blading may extend completely around the circumference, with no omission for blank closing pieces. (See fig. 49.)

The manganese-copper blades are retained in dovetailed grooves, as shown in fig. 50. Each blade has one end upset and a small projecting foot formed, which hooks under a wedge-shaped packing piece fitting the groove, and inserted between each successive blade in a row. The nickel-steel reaction blades are secured in double-tongue grooves with double wedges.

All reaction blades are fastened together in sections

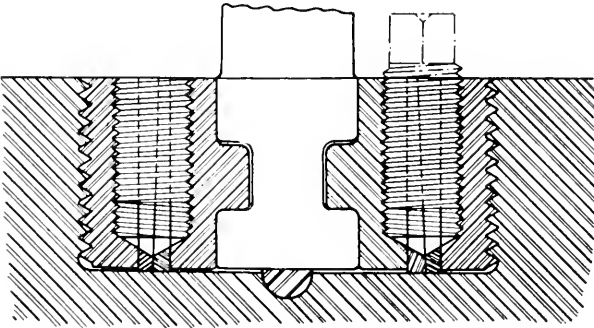
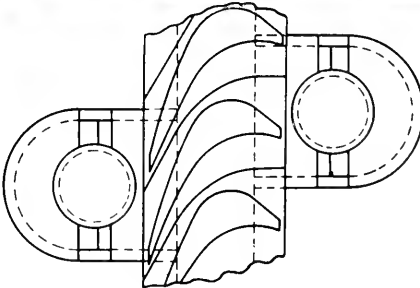
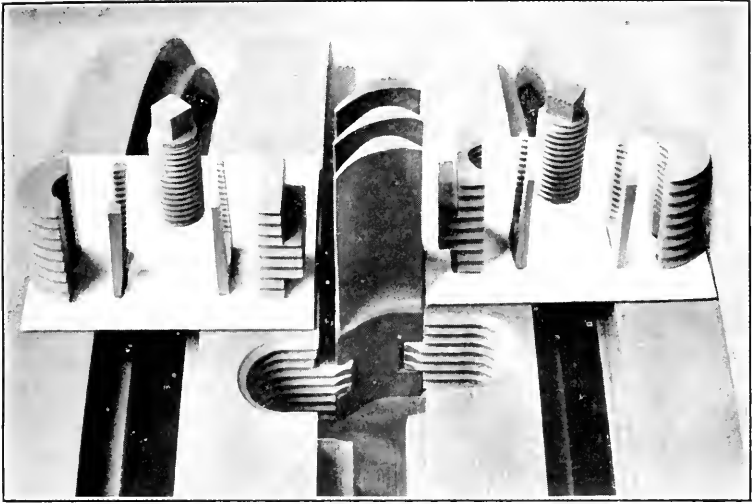


FIG. 49. Blade-Locking Device.

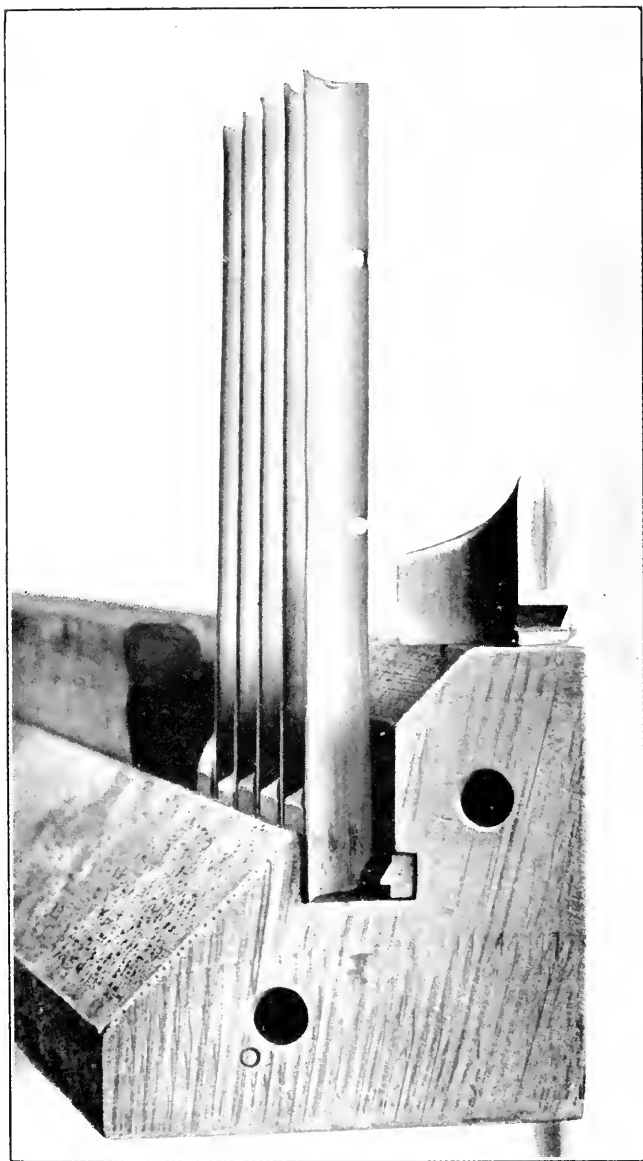


FIG. 50. Showing Method of Fixing Reaction Blading.

PLATE VII.

by a round wire inserted through a hole in each blade, and soldered thereto.

The minimum clearance between the tips of the moving blades and the cylinder is 0.028 inch, which occurs between the tip of the shortest moving blade and the cylinder. This clearance is gradually increased until a maximum of 0.096 inch is obtained between the last stationary blade and the rotor. While it is advisable to keep the space through which steam may leak from one stage to the next as small as possible, clearances must be made sufficiently large to ensure thorough reliability against blade rubs. However, as a precaution against serious damage, the thickness at the tip of each blade is reduced to a thin edge. Should contact occur, these fine edges will wear down or burr over without other important consequences.

The Governor.—The governor is of the hydraulic oil impeller type. It is operated by the variation in oil pressure produced by changes in speed of an impeller oil-pump, (seen in fig. 46,) mounted on the thrust end of the turbine rotor. With an increase in load on the turbine and consequent decrease in speed, the oil pressure decreases as the square of the change of speed and the governor opens the inlet valves to admit more steam to the turbine. The converse is true in the case of a decrease in load and an increase in speed.

Oil at impeller discharge pressure operates directly upon the under side of a pilot-valve piston, and is counter-balanced by a spring. Any increase or decrease in oil pressure will allow the pilot valve to open, discharging or admitting oil over the operating piston, which in turn moves up or down and closes or opens the steam admission valves. This type of governor has but few moving parts to wear and is extremely sensitive in its operation.

Protection against damage to the turbine through failure of the oil supply is amply provided for by a trip piston, which comes into operation should the bearing pressure of the oil be reduced below 3 pounds per square inch. This piston relieves the oil pressure from the

operating piston, rendering it inoperative, in which case the heavy springs on the valves force them shut.

As a precaution against overspeeding of the turbine, there is provided an automatic safety stop governor, mounted on the end of the turbine shaft, by means of which the throttle valve is tripped shut if the turbine speed for any reason exceeds a predetermined limit.

Lubrication.—Oil is drawn from the reservoir by means of the rotary pump mounted on the turbine spindle, and delivered to the governor mechanism. The oil not used by the latter is piped to an oil ejector, where its pressure is reduced to that of the bearing supply line, and at the same time it entrains more oil from the reservoir. The ejector is so proportioned that it will deliver about 2 gallons of low-pressure for each gallon of high-pressure oil supplied. From the ejector the oil passes through the cooler on its way to the impeller suction and Kingsbury thrust bearing. After passing through the thrust bearing it is delivered to the shaft bearings, from which it is drained back to the oil strainer and reservoir. Oil used in the governor mechanism is drained directly to the oil strainer and reservoir. The oil cooler is of the surface type, manufactured by the Griscom-Russell Company. The oil strainer contains three screens, each of different degrees of fineness. The first screen is a perforated plate, the second a thirty mesh to the inch screen, and the third a seventy mesh to the inch screen. Each screen is held in place by guides, and since they are three in number they can be removed for cleaning, one at a time.

The oil flows from the strainer to the reservoir, which is a cast-iron box located at the governor end of the turbine.

The whole oil system of the machine requires about 140 U.S. standard gallons to fill it.

A steam-driven auxiliary oil pump, with governor control, is furnished for use when starting and stopping the unit. In starting the turbine the auxiliary pump furnishes oil to the governor, impeller, ejector, and bearings. When the turbine reaches full speed this pump

is automatically stopped. Should, however, the bearing pressure be reduced below a predetermined amount, the automatic control functions, starting the auxiliary pump and ensuring a supply of oil to the bearings.

Main Bearings.—These bearings consist of a cast-iron shell, horizontally split and lined with babbitt.

Each bearing is supported in a bored support ring in the pedestal by means of four steel blocks or bearing keys.

Hard sheet-metal liners of various thicknesses are placed under each key. By shifting these liners from top to bottom or from side to side, the rotor is accurately located in the centre of the cylinder.

Glands.—The sealing glands at both the governor and coupling ends of the turbine are a combination of the ordinary labyrinth steam-seal and a water-seal type. The principle of the water-seal is that a small impeller mounted on the turbine shaft, and having curved vanes and a central diaphragm in the plane of rotation, revolves in a chamber in the casing. When the machine is running at normal speed water is thrown to the periphery of the wheel, and the escape of steam past the impeller effectively prevented.

TOPICS FOR DISCUSSION

1. The overall efficiency of the modern steam-power plant is in the neighbourhood of 15 per cent. Point out the thermodynamic reason for this comparatively low efficiency, and explain by reference to the temperature-entropy chart the advantages of the use of mercury in association with steam as the working substance in a binary vapour prime-mover.

2. Discuss the various points of departure of the actual from the ideal indicator diagram for a steam-engine. Indicate those measures which have been taken to minimise the loss arising from cylinder condensation, and tell what effect these have had upon the general trend of steam-engine design.

3. Explain how a temperature-entropy diagram for the cycle of the steam-engine may be drawn for a given indicator card.

How may an adiabatic for the cushion steam alone be plotted on the chart? Does the area to the left of (v) (i) in fig. 28 represent lost work in the same sense as the area to the right of (ii) (iii) (iv) when the actual cycle is compared with the ideal Rankine cycle?

4. Make a diagrammatic cross-section of the Skinner 'Unaflo' engine and describe its action. What are the advantages of a poppet valve? What is meant by a 'balanced' valve?

In recent designs of this engine, the poppet valve consists of two parts, an upper and a lower, flexibly connected and made steam-tight by a joint of the ordinary split-ring type. What purpose is served by this arrangement?

5. Design a suitable apparatus for determining the twist of a propeller shaft over a given length. Having found this quantity by observation, show how the horse-power transmitted by the shaft may be estimated if the shear modulus of the material of the shaft and the number of revolutions per minute are known.

6. Derive the formula giving the mass of the discharge from a convergent-divergent nozzle of known throat area, when the initial pressure of dry and saturated steam is given. Show that if supersaturation be presumed to persist as far as the throat, the discharge under these circumstances is about 5 per cent. in excess of that calculated on the assumption of stable and frictionless flow.

7. Show by reference to the pressure-volume diagram that the drop in total heat of steam flowing through a suitably proportioned nozzle is a measure of the gain in kinetic energy, friction being neglected.

8. Find an expression for the work done upon a given turbine blade ring per pound of steam flowing, in terms of the velocities of whirl at entrance and exit and the velocities of the tips of the blades at entry and exit, supposing these to differ as in the case of the radial flow machine of the Ljungström type.

9. Discuss the thermodynamic advantages of heating the feed by bleeding the turbine at a number of points throughout the range of the expansion.

10. Show by means of a diagram the action of the water seal.

WORKED EXAMPLES

1. Calculate as accurately as the steam tables permit the final dryness fraction of steam which has expanded adiabatically from an initial state of 100 pounds per square inch and 392 deg. Fahr., to a pressure of 15 pounds per square inch.

The specific heat of steam at 100 pounds per square inch is 0.55.

If this expansion be represented by the equation

$$PV^n = \text{const.}$$

what is the value of 'n' satisfying initial and final conditions? [B.Sc. (Eng.) Lond.]

During adiabatic expansion the entropy remains constant.

From the steam tables, $\phi_{s100} = 1.6020$.

Now the temperature of saturation for this pressure is 327.8 deg. Fahr. Hence there are $392 - 327.8$ or 64.2 deg. superheat.

$$\begin{aligned}\phi_{sup100} &= \phi_{s100} + 2.303 \times 0.55 \times \log \frac{852}{787.7} \\ &= 1.6020 + 0.431 \\ &= 1.6451.\end{aligned}$$

The dryness after adiabatic expansion is given by

$$\begin{aligned}&\frac{\phi_{sup100} - \phi_{w15}}{\phi_{s15} - \phi_{w15}} \\ &= \frac{1.6451 - 0.3133}{1.4416} \\ &= 92.38 \text{ per cent.}\end{aligned}$$

Required next, to find that value of 'n' in the expression

$$PV^n = \text{const.}$$

which satisfies both initial and final conditions.

We have

$$P_1 V_1^n = P_2 V_2^n$$

$$\therefore \log P_1 + n \log V_1 = \log P_2 + n \log V_2$$

$$\therefore n = \frac{\log P_2 - \log P_1}{\log V_1 - \log V_2}$$

$$V_1 = 0.9238 \times 26.27$$

$$= 24.26 \text{ c.f.}$$

$$V_2 = 4.89 \text{ c.f. by interpolation}$$

$$n = \frac{2 - 1.1761}{1.3849 - 0.6893}$$

$$= 1.184.$$

2. A single-cylinder double-acting steam-engine has a piston diameter of 11 inches, a stroke of 18 inches, and the clearance volume is 8.5 per cent. of the swept volume. The steam supply is at 70 pounds per square inch gauge pressure and the exhaust at atmospheric pressure (30 inches of mercury). Cut-off is at 40 per cent. of the stroke and compression at 80 per cent.

Assuming that the actual indicator card has an area of 90 per cent. of the theoretical card having hyperbolic expansion and compression, find the indicated horsepower of the engine when running at 185 revolutions per minute. [B.Sc. (Eng.) Lond.]

Let P_{in} = pressure during admission in pounds/sq. ft.

„ P_{ex} = „ at release „ „ „

„ P_b = back pressure „ „ „

„ V_{cl} = volume of the clearance in cubic feet

„ V_{co} = „ at cut-off „ „ „

„ V_{end} = „ at end of stroke

„ V_{com} = „ at commencement of compression.

The work done per stroke

$$= P_{in} (V_{co} - V_{cl}) + \int_{V_{co}}^{V_{end}} P dV$$

$$- P_b (V_{end} - V_{com}) - \int_{V_{cl}}^{V_{com}} P dV.$$

Volume swept through by piston

$$= \frac{\pi d^2}{4} l = \frac{3 \cdot 14 \times (\frac{11}{12})^2}{4} \times 1 \cdot 5$$

$$= 0 \cdot 9893 \text{ cu. ft.}$$

$$V_{cl} = 0 \cdot 9893 \times 0 \cdot 085$$

$$= 0 \cdot 0841 \text{ cu. ft.}$$

$$V_{end} = 0 \cdot 9893 + 0 \cdot 0841$$

$$= 1 \cdot 0734 \text{ cu. ft.}$$

$$V_{com} = 0 \cdot 9893 \times 0 \cdot 2 + 0 \cdot 0841$$

$$= 0 \cdot 2820 \text{ cu. ft.}$$

$$V_{co} = 0 \cdot 9893 \times 0 \cdot 4 + 0 \cdot 0841$$

$$= 0 \cdot 4798 \text{ cu. ft.}$$

$$\text{Work done} = 84 \cdot 7 \times 144 (0 \cdot 4798 - 0 \cdot 0841) \dots (i)$$

$$+ 84 \cdot 7 \times 144 \times 0 \cdot 4798 \times 2 \cdot 303 \times \log \frac{1 \cdot 0734}{0 \cdot 4798} \dots (ii)$$

$$- 14 \cdot 7 \times 14 \cdot 4 (1 \cdot 0734 - 0 \cdot 2820) \dots (iii)$$

$$- 14 \cdot 7 \times 144 \times 0 \cdot 2820 \times 2 \cdot 303 \times \log \frac{0 \cdot 2820}{0 \cdot 0841} \dots (iv)$$

$$= 4827 + 4712 - 1675 - 722 \cdot 3$$

$$= 7141 \cdot 7 \text{ ft.-pounds.}$$

Actual work done per stroke

$$= 0 \cdot 90 \times 7141 \cdot 7$$

$$= 6427 \cdot 5 \text{ ft.-pounds.}$$

$$\text{Horse-power} = \frac{2 \times 6427 \cdot 5 \times 185}{33000}$$

$$= 72 \cdot 06$$

3. A convergent-divergent nozzle is to be designed to discharge 0·15 pound of steam per second into a vessel in which the pressure is 20 pounds per square inch when the nozzle is supplied with steam at 100 pounds per square inch superheated to 392 deg. Fahr. Find the throat and

exit diameters of the nozzle on the assumption that the loss in the diverging part is 10 per cent. of the total heat drop. [B.Sc. (Eng.) Lond.]

By reference to the Mollier chart of entropy and total heat we observe that the early stages of expansion lie within the region of superheat, and that the state point crosses the boundary of the normally wet region in the neighbourhood of the throat.

The heat drop to the throat on the assumption of frictionless, adiabatic flow is found to be about 48 B.Th.U., giving a velocity of

$$\begin{aligned} v_t &= \sqrt{2gJ \, 48} \\ &= 1550 \text{ ft./sec.} \end{aligned}$$

The specific volume at the throat on the assumption of equilibrium expansion is 7.44 cubic feet, and for the mass flowing per second,

$$V_t = 7.44 \times 0.15 = 1.116 \text{ cubic feet.}$$

Hence, area of throat required is

$$\begin{aligned} A_t &= \frac{V_t}{v_t} \times 144 \text{ square inches} \\ &= \frac{1.116 \times 144}{1550} \\ &= 0.104 \text{ square inch.} \end{aligned}$$

The total isentropic heat drop to the back pressure measured on the chart is 124 B.Th.U. The friction loss is therefore $0.10 \times 124 = 12.4$ B.Th.U., and the net heat drop is equal to 111.6 B.Th.U.

$$\begin{aligned} \text{Hence the velocity at exit} &= \sqrt{2gJ \, 111.6} \\ &= 2363 \text{ ft./sec.} \end{aligned}$$

$$\begin{aligned} V_{ex} &= q_{ex} V_{sp} m = 0.954 \times 20.08 \times 0.15 \\ &= 19.15 \text{ cu. ft.} \end{aligned}$$

$$\begin{aligned} \text{Area} &= \frac{19.2 \times 144 \times 0.15}{2360} \\ &= 0.175 \text{ square inch.} \end{aligned}$$

4. Steam issues from a nozzle on to the blade ring of an impulse steam-turbine with a velocity of 1500 feet per second. The receiving and discharging tips of the moving blades are inclined at 30 degrees to the plane of motion and the nozzle is inclined at 20 degrees to that plane. The mean diameter of the blade-ring circle is 3 feet. The discharge from the nozzle is 0·23 pound per second. Assuming a friction loss of 15 per cent. in the velocity of the steam relative to the blade whilst passing through the blade ring, find (1) the instantaneous torque on the shaft when the turbine is beginning to move; (2) the speed of the turbine so that the steam shall discharge axially, and the horse-power developed at this speed. [B.Sc. (Eng.) Lond.]

(1) Required to find the instantaneous torque on the shaft when the turbine is beginning to move.

P, the thrust upon the blade in the direction of motion, is equal to $\frac{m}{g} \{v_{wcn} - v_{wex}\}$

$$\begin{aligned} T, \text{ the resulting torque,} &= \frac{mr}{g} \{v_{wcn} - v_{wex}\} \\ &= \frac{mr}{g} \{v_{cn} \cos \alpha - v_{ex} \cos \alpha_1\} \end{aligned}$$

Now, since the blade is at rest at the instant under consideration,

$$\begin{aligned} v_{cn} &= v_{rcn} \\ \text{and } v_{ex} &= v_{rcx} \end{aligned}$$

We are given that

$$v_{ex} = kv_{cn}$$

where 'k' is the given coefficient relating the velocity at exit to the velocity at entrance.

Hence

$$T = \frac{0.23 \times 18}{32.2} \{1500 \cos 20^\circ + 0.85 \times 1500 \cos 30^\circ\}$$

$$= \frac{0.23 \times 18}{32.2} \left\{ 1500 \times 0.9397 + 0.85 \times 1500 \times 0.866 \right\}$$

$\therefore T = 323.4$ pounds-inches.

(2) Next, to determine V_b in order that the steam may leave the wheel axially,

$$\frac{v_b}{v_{relx}} = \cos \beta_1$$

and

$$v_{relx} = kv_{reln}$$

$$v_{reln}^2 = v_{en}^2 \sin^2 \alpha + \{v_{en} \cos \alpha - v_b\}^2$$

also

$$\frac{v_b}{kv_{reln}} = \cos \beta_1$$

$$\therefore v_b = k \cos \beta_1 \{v_{en}^2 \sin^2 \alpha + (v_{en} \cos \alpha - v_b)^2\}^{\frac{1}{2}}$$

$$\text{and } v_b^2 = k^2 \cos^2 \beta_1 \{v_{en}^2 \sin^2 \alpha + (v_{en} \cos \alpha - v_b)^2\}$$

$$= k^2 \cos^2 \beta_1 v_{en}^2 - k^2 \cos^2 \beta_1 2v_{en} \cos \alpha v_b$$

$$+ k^2 \cos^2 \beta v_b^2$$

$$\therefore 0.4572 v_b^2 + 1527 v_b - 1,219,000 = 0$$

$$v_b = \frac{-1527 \pm \sqrt{1527^2 + 4 \times 0.4572 \times 1,219,000}}{0.9144}$$

$$= 664.9 \text{ feet/second.}$$

In order to determine the horse-power developed at this speed we have

$$\text{H.P.} = \frac{PV_b}{550}$$

$$= \frac{m}{g} \frac{\{v_{wcn} - v_{wex}\} v_b}{550}$$

Now

$$v_{wex} = 0$$

and

$$v_{wcn} = 1500 \cos \alpha = 1409 \text{ ft./sec.}$$

$$\text{H.P.} = \frac{0.23 \times 1409 \times 664.9}{31.17 \times 550}$$

$$= 12.57.$$

EXAMPLES FOR PRACTICE

1. The following data were obtained in a steam boiler trial of five hours' duration : Coal used, 1390 pounds ; carbon content of fuel, 0.886 per cent. ; lower calorific value of coal, 14,800 B.Th.U. per pound ; water evaporated, 12,175 pounds ; feed temperature, 110 deg. Fahr. ; boiler-gauge pressure, 50.2 pounds per square inch ; barometer, 30.4 inches ; volumetric analysis of dry flue gas, CO_2 10.2, O_2 8.6, N_2 81.2 ; flue gas temperature, 533 deg. Fahr. Calculate from these data the efficiency of the boiler and find what percentage of the calorific value of the fuel is carried away by dry flue gas (specific heat 0.24). [B.Sc. (Eng.) Lond.]

(In working this problem note that the weight of dry flue gas per pound of carbon in the coal

$$\begin{aligned}
 &= \frac{\text{relative weight of dry flue gas}}{\text{relative weight of carbon in gas}} \\
 &= \frac{44\text{CO}_2 + 28\text{N}_2 + 32\text{O}_2}{12\text{CO}_2}
 \end{aligned}$$

2. The composition by weight of the coal used to fire a boiler is carbon, 0.863 ; hydrogen, 0.038 ; oxygen, 0.004 ; ash, 0.095. Assuming that the ratio of air used to that theoretically required for combustion is 1.6, find the weight of flue gas per pound of coal fired.

If the chimney draught is one inch of water and the temperature of the flue gas at the base of the chimney is 752 deg. Fahr., what should be the internal cross-sectional area of the chimney if the boiler is to consume $1\frac{1}{2}$ tons of coal per hour ? Assume that one-tenth of the draught is available for giving velocity, the rest of the draught being used in overcoming friction of the air passing through the fuel bed. [B.Sc. (Eng.) Lond.]

(Weight of flue gas per pound of coal

$$= 11.6\text{C} + 34.8\left(\text{H} - \frac{\text{O}}{8}\right) + 1$$

in which C, H, and O are the weights of carbon, hydrogen, and oxygen per pound of coal.

Note also that

$$\frac{P_a V_a}{P_{fg} V_{fg}} = \frac{n}{n+1} \frac{T_a}{T_{fg}},$$

' n ' being the number of pounds of air which have taken up one pound of fuel in passing through the furnace.

In computing the necessary chimney area, R for air may be taken as 53.4 .)

3. The effective volume of the cylinder of a steam-engine is 0.92 cubic foot and the clearance volume is 8.5 per cent. of this volume. The weight of steam in the cylinder at half-stroke is 0.063 pound and the pressure is 42 pounds per square inch. The steam expands hyperbolically to 0.95 stroke. Calculate the final dryness of the steam, and determine the number of heat units that pass through the walls into the cylinder during expansion. [B.Sc. (Eng.) Lond.]

4. A compound steam-engine is to develop 125 horsepower at 110 revolutions per minute. The steam supply is at 105 pounds per square inch, and the condenser pressure is 3 pounds per square inch. Assuming hyperbolic expansion, an expansion ratio of 15 , a diagram factor of 0.7 , and neglecting clearance volume and receiver losses, determine the diameters of the cylinders so that they may develop equal powers.

The stroke of each piston may be taken equal to the low-pressure cylinder diameter. [B.Sc. (Eng.) Lond.]

5. The following observations were made during a trial of a double-acting steam-engine: Piston diameter, $15\frac{1}{2}$ inches; stroke, 18 inches; revolutions per minute, 158 ; torque on brake shaft, 1900 pounds feet; steam supply, 16.03 pounds per minute; gauge pressure, 150 pounds per square inch; vacuum, 24.1 inches; barometer, 29.2 inches; area of indicator card, 0.96 square inch; length of card, 3.3 inches; indicator spring, 100 ; cooling water per minute, 431.0 pounds; rise in temperature of cooling water, 32.9 deg. Fahr.; temperature of condensate, 118.6 deg. Fahr.

Draw up a heat balance-sheet for the engine per minute, reckoning the heat supplied from water at the temperature corresponding to the condenser pressure, and find the mechanical and thermal efficiencies of the engine. [B.Sc. (Eng.) Lond.]

6. A four-cylinder compound locomotive works at a boiler pressure of 200 pounds per square inch, by gauge, and exhausts to the atmosphere. All cylinders have the same stroke, namely, 26 inch, and the diameter of the two high-pressure is 14 inch and of the two low-pressure 23 inch. Cut-off occurs in the high at 0.5 of the stroke and at 0.6 of the stroke in the low-pressure cylinder. Assuming hyperbolic expansion and a diagram factor of 0.65, estimate the horse-power developed when running at a speed of 30 miles per hour.

The driving wheels are 6 feet 8 inches in diameter. Neglect clearance and take atmospheric pressure at 15 pounds per square inch. [B.Sc. (Eng.) Lond.]

7. Explain why the increase in efficiency obtained by using superheated steam in reciprocating engines is more than the gain which would be expected from the increase in efficiency of the corresponding Rankine cycle. A steam-engine working over a constant pressure range of 90 to 14.7 pounds per square inch has a constant ratio of expansion, and is supplied with steam of various degrees of superheat. The steam consumption was observed to be as follows :

Temperature of steam supply, Fahr., dry 395, 474, 554.
Steam per I.H.P. hour, 42.0, 37.4, 32.3, 27.1.

Plot curves showing the variation with degrees of superheat of the actual and Rankine efficiencies with complete expansion. [B.Sc. (Eng.) Lond.]

8. The pressure indicated at cut-off in the cylinder of a jacketed steam-engine is 75 pounds per square inch, the volume is 0.45 cubic feet and the dryness fraction of the steam is 0.73.

If the values of pressure and volume at release are 34.2 and 1.05, find the heat which passes through the cylinder walls during expansion. [B.Sc. (Eng.) Lond.]

9. In a reaction steam-turbine the fixed and moving blades have inclinations at the receiving and discharging tips of 35 deg. and 20 deg. respectively. If the steam passes through the blades without shock, what would be the work developed in a ring of moving blades per pound of steam if the mean diameter of the blade-ring circle is 5 feet 6 inches and the speed of rotation 700 revolutions per minute ?

Assuming an efficiency of 85 per cent. for the one pair of fixed and moving blades, find the heat drop which takes place in the pair, and find the state of the steam when arriving at the fixed blades if the steam has a pressure of 20 pounds per square inch and dryness 0.92 when leaving the moving blades. [B.Sc. (Eng.) Lond.]

10. Dry and saturated steam is allowed to discharge through a short convergent nozzle, from a pressure of 25 pounds per square inch abs., into the atmosphere. If the diameter at exit is 0.5 inch, find the mass of the discharge in pounds per hour (*a*) on the assumption of stable isentropic flow, (*b*) on the assumption that the steam is supersaturated at exit.

11. The exhaust from a steam-engine is to be utilized in a reaction-turbine which is to run at 300 r.p.m. The engine uses 1950 pounds of steam per hour, and the steam is supplied to the turbine at a pressure of 20 pounds per square inch and has a dryness 0.95. If the guide blades in the first ring in the turbine discharge at 20 degrees to the plane of motion and the receiving tips of the moving blades are inclined 35 degrees to this plane, and if the blade height is one-fifteenth of the mean blade-ring diameter, find this diameter so that the steam shall pass on to the moving blades without shock, and, neglecting friction, find the power developed in the first moving ring of blades which have the same discharge angle as the fixed blades. [B.Sc. (Eng.) Lond.]

(Note that the product of the axial component of the velocity of flow and the area comprised between the blade tip and root circles is equal to the volume of steam passing per unit time.)

12. The first stage of an impulse turbine is supplied with steam at 200 pounds per square inch, superheated to 482 deg. Fahr.

The velocity is compounded, there being two rings of moving blades, separated by a ring of fixed blades. The pressure in the wheel chamber is 100 pounds per square inch.

Assuming an 8 per cent. loss in the nozzle, find the velocity of the steam as it leaves the nozzle. Given that the angle of the tips of the moving blades is 30 deg., and that the blade velocity is one-fifth of that of the steam discharging from the nozzle, find the work done per pound of steam if the velocity of the steam relative to the blade is reduced by 12 per cent. in its passage through each blade ring, and if the nozzle and fixed blade angles are designed so that steam shall pass through without shock. Determine also the efficiency of the stage. [B.Sc. (Eng.) Lond.]

13. A stage in an impulse steam-turbine consists of converging nozzles and one ring of moving blades. The nozzle angles are 22 degrees and the moving blades have both blade-tip angles of 35 degrees. If the velocity of the steam at the exit from the nozzle is 1500 feet per second, find the blade speed so that the steam shall pass on to the blade without shock, and find the stage efficiency, neglecting frictional losses, if the blades run at this speed.

If the relative velocity of steam to blade is reduced by 15 per cent. in passing through the blade ring, find the actual efficiency and the end thrust on the shaft when the blade ring develops 50 horse-power. [B.Sc. (Eng.) Lond.]

(The terms 'stage efficiency' and 'actual efficiency' refer to the ratio between the work absorbed by the rotor and the initial kinetic energy of the jet in each case.)

14. In a reaction steam-turbine the blade tips are inclined 35 degrees and 20 degrees to the direction of motion.

The guide blades are of the same shape as the moving blades but reversed in direction. At a certain place in the turbine the drum diameter is 3 feet 6 inches and the blades are 4 inches high, and at this place the steam has a

pressure of 25 pounds per square inch and dryness 0.935. If the speed of the turbine is 250 revolutions per minute and the steam passes through the blades without shock, find the power developed in the ring of moving blades. [B.Sc. (Eng.) Lond.]

(Note that the speed of the turbine fixes the velocity of the blades, and hence also the velocity at entrance for no shock. The axial component of the velocity of flow may then be found, and the volume of steam flowing per second is then readily determined from a knowledge of the area of the annular space through which it flows.

Now find the mass flowing per second, and hence the power developed.)

15. Determine the twist in degrees in a length of nine feet of a propeller shaft 18 inches in diameter and 8 inches internal diameter if the maximum shear stress does not exceed 8000 pounds per square inch. What horse-power would be transmitted when rotating at 110 revolutions per minute?

SECTION III

THE INTERNAL COMBUSTION ENGINE—
THE DIESEL ENGINE—THE GAS-TUR-
BINE—THE EXHAUST-GAS-TURBINE—
MECHANICAL REFRIGERATION, IN-
CLUDING THE PRODUCTION OF VERY
LOW TEMPERATURES

THE INTERNAL COMBUSTION ENGINE

IN the derivation of power from heat there are, broadly, two major divisions of the general method. Either the products of combustion may give up heat to some other substance, which by expansion will cause work to be done, as in the case of the steam prime-mover, or the heated gases resulting from the combustion of the fuel may themselves be used as the vehicle by which the desired transformation is effected.

Action of the Internal Combustion Engine.—The latter plan is followed in the large class of prime-mover, which combines the function of the furnace, boiler and engine of the steam plant in a single organ. The characteristic action of the internal combustion engine consists in filling the cylinder with air, which may, or may not, be charged with fuel at the commencement of the cycle. In order to increase the availability of the heat about to be added on ignition of the charge, the mixture is first compressed. At some predetermined point at or near the end of the stroke of the piston inflammation occurs, and according to whether the action is sudden (thus partaking of the nature of an explosion as in the case of the automobile engine, in which the fuel enters with the induction air) or comparatively slow (as in the case of the Diesel, in which a stream of fuel enters the cylinder during the early stages of the working stroke), we obtain the general classification of constant volume and constant pressure cycles.

Comparison of Actual and Ideal Cycles.—As has been already shown, these cycles depart from the ideal

of Carnot in two important respects, namely, heat is taken in while the temperature of the working substance rises and heat is rejected as it falls. In order, therefore, to form some estimate of the performance of actual engines, comparison is made with an ideal engine following the appropriate standard cycle and using air, to which, for this purpose, are attributed the properties of a perfect gas. In both the 'constant volume' and 'constant pressure' cycles the efficiency was found

to be
$$1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

r being equal to the ratio of compression and expansion in each case. These efficiencies are always less than the corresponding Carnot efficiencies calculated for the temperature range over which the engine works.

Influence of the Imperfection of the Gas upon the Efficiency.—By perfection in a gas we understand obedience to the law

$$\frac{PV}{T} = \text{constant},$$

and further that the internal energy of the gas consists of energy of translation and energy of rotation, only. This latter assumption involves constancy of specific heat. Now, while it may be shown that in the case of the actual gas the whole of the communicable energy of translation and rotation is equally divided amongst the various effective degrees of freedom of the molecule and that the absolute temperature of the gas is proportional to this quantity, in polyatomic gases a certain proportion of the whole internal energy exists as vibrational energy of the component parts of the molecule or, possibly, as potential energy of separation as the rotational energy increases, and in all cases, but to a much less degree, as vibrational energy of the constituents of the atoms. If the ratio which this latter quantity bore to the whole internal energy remained constant, the rate of increase of internal energy of the gas with respect to temperature would be a constant and thus the specific heat at constant volume would be a constant. As a

matter of fact, observation shows that the ratio of vibrational energy to the total internal energy increases as the temperature rises, and while the gas may, and in general the permanent gases do, obey the general law more or less exactly, there is an increase in the specific heat at constant volume with increase in temperature. If, then, the permanent gases, and in particular the mixture we commonly find in the cylinder of the engine, depart but little from the laws of Boyle and Joule, it follows that R is a constant, and therefore, since

$$C_p - C_v = R \text{ (in work units),}$$

if C_v varies with the temperature, so also C_p varies in such a manner as to maintain their difference constant.

The effect of such variation is that neither the pressures nor the temperatures expected upon the assumption of constancy of specific heat are actually realized and the efficiency is reduced.

Actual Values of the Specific Heat for a Monatomic Gas.—As might be inferred from the constitution of a monatomic gas, little variation of the specific heat is observed.

In any given case, such as mercury vapour, C_p and C_v may be determined as follows, when the molecular weight is known.

According to Avogadro's law, equal volumes of different gases at the same temperature and pressure have the same number of molecules, and therefore also the weights of equal volumes under standard conditions are proportional to the molecular weights. If, then, the

equation
$$\frac{PV}{T} = \text{constant}$$

be written per a mass of the gas in each case, numerically equal to the molecular weight (a quantity which we may call a 'pound molecule'), thus

$$PVm = mRT,$$

it follows that since $mV = \text{constant}$ for different gases when P and T are constant, so also is the quantity mR .

This is known as the universal gas constant and may be written as R' to differentiate it from R , the constant having reference to unit mass. When P is measured in pounds per square foot, volume in cubic feet, and temperature in degrees Centigrade abs., R' assumes the value 2779 foot-pounds per pound molecule. In pound calories—that is, the heat necessary to raise the temperature of one pound of water one-hundredth of the temperature range, freezing-point to boiling-point, under atmospheric pressure—this becomes

$$\frac{2779}{1400} = 1.985.$$

From this observed fact, and knowing that, according to the molecular kinetic theory, γ for a monatomic gas is 1.667, we have

$$C'_p - C'_v = 1.985,$$

$$\frac{C'_p}{C'_v} = 1.667,$$

whence

$$C'_p = 4.962$$

and

$$C'_v = 2.997.$$

Since there is little, if any, departure in practice from the assumed conditions, these values may be taken as exact.

Actual Values of the Specific Heat for a Diatomic Gas.—If in the case of a diatomic gas the simple assumed conditions actually existed, we should have

$$C'_p - C'_v = 1.985,$$

$$\frac{C'_p}{C'_v} = 1.4,$$

whence

$$C'_v = 4.962$$

and

$$C'_p = 6.947.$$

If these results be compared with the values found experimentally for air, nitrogen, oxygen, and carbon monoxide, they are found to agree quite closely at temperatures in the neighbourhood of the freezing-point of

water. For instance, the apparent molecular weight of air being

$$\frac{79.1 \times 28}{100} + \frac{20.9 \times 32}{100} = 28.85,$$

C_v for air would therefore be $\frac{4.962}{28.85} = 0.172$ per pound.

Measured values of this quantity for air at freezing-point and atmospheric pressure give 0.1727. This indicates that under these conditions but a small proportion of the internal energy is in the form of vibrational energy.

As, however, the temperature rises, an expression of the form

$$C'_v = 4.98 + 0.001t$$

(t being in degrees Centigrade), in pound calories per pound molecule, becomes more nearly applicable.

This may also be written

$$C'_v = 4.98 + (0.001T - 0.001 \times 273)$$

or
$$C'_v = 4.707 \times 0.001T$$

(T being in degrees Centigrade abs.).

This expression will answer for moderate temperature ranges, but careful experiments appear to indicate that an expression which includes a third term, involving the second power of the temperature, should be used for high temperatures.

Experimental Methods of Determining the Specific Heat.—When a gas expands, or is compressed adiabatically, the area under the curve relating pressure and volume gives a measure of the change in internal energy. If, then, a gas of known composition be selected and subjected to alternate compression and expansion in the cylinder of an engine, under such circumstances that the loss of heat to the cylinder walls is negligible or may be estimated, the change of internal energy between assigned limits, say, of volume, may be determined. The relationship which exists between the change of internal energy and the corresponding change

of temperature is then found by the application of the law

$$\frac{PV}{T} = \text{constant}$$

to each limiting set of conditions. Now, since

$$C_v dT = dE,$$

$$\therefore C_v = \frac{dE}{dT}.$$

Hence, when an expression has been found which relates the internal energy to the temperature, the rate of change of E with respect to T at any point gives a measure of the specific heat at constant volume at that point. As an illustration of this method, Sir J. A. Ewing quotes the results given in the first report of the British Association Committee on Gaseous Explosions. Partly by the method above outlined, the following expression was derived for a mixture consisting of 5 per cent. CO_2 , 12 per cent. H_2O , and 83 per cent. N_2 , and surplus O_2 ,

$$E' = 5 \cdot 2t + 0 \cdot 00043t^2 + 0 \cdot 0000002t^3,$$

t being in degrees Centigrade and E' in pound calories per pound molecule.

On differentiation we have

$$C'_v = 5 \cdot 2 + 0 \cdot 00086t + 0 \cdot 0000006t^2.$$

Compare this result for $t = 0$ with the figure obtained for the ideal diatomic gas, namely, $C'_v = 4 \cdot 962$.

Values for the Specific Heats of the Triatomic Gases.—By a similar calculation to the one above,

since

$$\gamma = 1 \cdot 333$$

and

$$C'_p - C'_v = 1 \cdot 985,$$

$$C'_v = 5 \cdot 961$$

and

$$C'_p = 7 \cdot 946.$$

Experiment shows, however, that even at moderate temperatures an appreciable part of the total internal energy consists of energy of vibration. Hence, in the

case of the triatomic gases the specific heats will be somewhat greater, while the specific heats of gases which are composed of more complex molecules are considerably higher than the above values.

Combustion of Gaseous Fuels.—In applying the general law

$$\frac{PV}{T} = \text{constant}$$

to the charge within the cylinder of an internal combustion engine, no account was taken of the influence of any change in the specific volume due to combustion. Were, for instance, contraction to occur, it is clear that if the amount is not negligible, its influence upon the calculated temperature may be observed from the equation

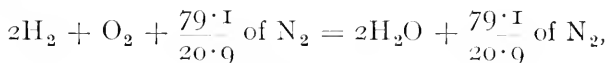
$$\frac{kP_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

or
$$T_2 = T_1 \frac{P_2V_2}{P_1V_1k},$$

in which P_1 , V_1 and T_1 relate to the conditions before explosion, P_2 , V_2 and T_2 after explosion, and k denotes the ratio of specific volume after combustion to the specific volume before combustion.

Consider, for instance, what happens when hydrogen is burned together with the minimum amount of air necessary for complete combustion.

We have



i.e. $2 + 1 + 3 \cdot 784$ give $2 + 3 \cdot 784$

or
$$k = \frac{5 \cdot 784}{6 \cdot 784} = 0 \cdot 8526.$$

Suppose 10 per cent. oxygen had appeared in the products of combustion. Let x cubic feet be the measure of the total exhaust volume. Then $0 \cdot 1 x = \text{O}_2$.

Associated with the surplus oxygen we should have had

$$\text{N}_2 = 0 \cdot 1 \times 3 \cdot 784x.$$

The balance, namely $(1 - 0.4784)x$, is water vapour and nitrogen in the ratio $\frac{2}{3.784}$.

Thus, expressing these quantities as percentages,

O ₂	10.00 per cent.
N ₂	37.84 „
H ₂ O $52.16 \times \frac{2}{5.784}$...	18.05 „
N ₂ $52.16 \times \frac{3.784}{5.784}$...	$\frac{34.15}{100.04}$ „

the original volume of the charge would have been

$$\left(\frac{52.16}{0.8526} + 47.84 \right) \text{ per cent.} = 109.01.$$

Hence $k = \frac{100}{109.01} = 0.9173.$

The following example will serve to illustrate a general case :

Composition of Gas.		Vol. of O ₂ reqd.	Products.		
			H ₂ O	CO ₂	N ₂
H ₂ 44 by $\frac{1}{2} = 22$	22	44	0	0
CH ₄ (Methane)	35 by 2 = 70	70	70	35	0
C ₂ H ₄ (Ethylene)	3 by 3 = 9	9	6	6	0
C ₄ H ₈ (Butylene)	2 by 6 = 12	12	8	8	0
CO 5 by $\frac{1}{2} = 2.5$	2.5	0	5	0
N ₂ 7	0	0	0	7
CO ₂ 4	0	0	4	0
	100	115.5	128	58	7

The original volume, with maximum oxygen utilization, is

$$100 + 115.5 \times \frac{100}{20.9} = 652.6.$$

Amount of contraction = $215.5 - 193 = 22.5.$

Hence the final volume = $652 - 22.5 = 629.5$

and thus k becomes $\frac{629.5}{652} = 96.8 \text{ per cent.}$

for full oxygen utilization. The actual contraction would, of course, be less than this on account of the air of dilution.

The Application of the Entropy Function to the Gas-Engine Cycle, taking account of the Variation of the Specific Heat.—If heat is added to a gas, the internal energy may be increased or work may be done, or both. Further, each of these quantities may be positive or negative according to whether heat is added or abstracted, or whether work is done by, or upon, the gas. The relationship which exists between those energy changes may be expressed by

$$\begin{aligned} dQ &= C'_v dT + \frac{PdV}{J} \\ &= (a_v + bT) dT + \frac{PdV}{J}. \end{aligned}$$

The corresponding change of entropy is given by

$$\begin{aligned} d\phi &= \frac{(a_v + bT) dT}{T} + \frac{PdV}{JT} \\ &= \frac{(a_v + bT) dT}{T} + \frac{R'dV}{JV}. \end{aligned}$$

Hence, when the temperature limits and corresponding limits of volume are assigned, we have

$$\phi - \phi_o = a_v \log_e \frac{T}{T_o} + b (T - T_o) + \frac{R'}{J} \log_e \frac{V}{V_o}.$$

This may also be written

$$\phi - \phi_o = a_p \log_e \frac{T}{T_o} + b (T - T_o) - \frac{R'}{J} \log_e \frac{P}{P_o}.$$

[Since

$$a_p = a_v + R'$$

and

$$\frac{P}{P_o} = \frac{TV_o}{T_o V}$$

and thus
$$\log_{\epsilon} \frac{P}{P_o} = \log_{\epsilon} \frac{T}{T_o} + \log_{\epsilon} \frac{V_o}{V}$$

$$= \log_{\epsilon} \frac{T}{T_o} - \log_{\epsilon} \frac{V}{V_o}.$$

Hence
$$-\log_{\epsilon} \frac{P}{P_o} = -\log_{\epsilon} \frac{T}{T_o} + \log_{\epsilon} \frac{V}{V_o},$$

whence the second expression for the change of entropy is obtained by substitution.]

If these symbols, namely,

$$C'_v = a_v + bT$$

and

$$C'_p = a_p + bT$$

have reference to a mass numerically equal to the molecular weight, the total change in entropy per unit mass may be found by dividing by the molecular weight.

In the practical application of the above equations, some uncertainty usually exists in the determination of the temperature at one point in the cycle; and this must be known or assumed before we can proceed. The reference point is commonly taken for the conditions at the commencement of compression where the temperature in most cases has a value ranging from 200 to 300 deg. Fahr. Given, then, the indicator card, a series of points are selected around the diagram and the temperature and volume or temperature and pressure ratios to zero conditions computed. The changes in entropy are then readily found.

THE DIESEL ENGINE

By way of exemplifying the fundamental principles upon which the internal combustion engine acts, the Diesel engine, and in particular the marine Diesel, will be selected for discussion.

Its comparatively recent success in large sizes and the inherent possibilities of greater development and wider application render it of particular interest.

The Cycle of the Diesel Engine.—The standard Diesel cycle is a modification of the ideal constant pressure cycle, heat being taken in at constant pressure but rejected at constant volume. There is a further classification under the terms 'four-stroke cycle,' and 'two-stroke cycle.'

In the former, air is drawn in during the induction stroke and compressed. Thereafter fuel oil is injected into the cylinder, commonly by high-pressure air blast, and ignites on encountering the compression heated cylinder air. During combustion the pressure remains sensibly constant. Thereafter there is a relatively long expansion, the diagram resembling that of the uniflow engine. On the opening of the exhaust valve the waste gases are discharged to the atmosphere and the cycle is completed by the return of the piston to its original position.

In the two-stroke cycle the induction and exhaust strokes are dispensed with, and these offices are performed by a scavenging blast which sweeps the cylinder clear of the waste products during the late part of the working stroke and early stages of the return.

The pressure limits between which large-scale Diesel engines work are fixed by experience and are normally

that of the atmosphere and about 500 pounds per square inch by gauge.

Maximum Temperature of Diesel Engine Cycle.—Since the temperatures reached within the cylinder of the Diesel are an important factor governing the design and influencing the development of the engine, it is of importance to inquire as to their order of magnitude and whether practical considerations impose limiting values which are lower than those which might be attained.

Let us assume that the temperature of the cylinder contents at the commencement of compression is 200 deg. Fahr., and further that the ratio of the pressure at the commencement of compression to the pressure at the end of compression is 35.

Now, in computing T_1 , the temperature at the end of compression, by means of the equation

$$T_1 = T_4 \left(\frac{P_1}{P_4} \right)^{\frac{\gamma-1}{\gamma}}$$

we are faced with a certain difficulty owing to the variation of γ , with change of the specific heat. We may, however, without important error, take a mean value of 1.38.

$$\begin{aligned} \text{Thus } T_1 \text{ becomes} &= 660 \times 35^{\frac{0.38}{1.38}} \\ &= 1757 \text{ deg. Fahr. abs.} \\ \text{or} &1297 \text{ deg. Fahr.} \end{aligned}$$

We come next to the problem of determining the maximum temperature which can possibly be reached at the end of combustion, with full oxygen utilization. A given quantity of air can only burn a limited amount of fuel. When the oxygen is all used up the action stops. As a matter of fact, a certain proportion of air of dilution must always be present to ensure complete combustion. Let us consider, first, how much air would be necessary for a fuel oil showing the following composition:

Carbon.....	86 per cent.
Hydrogen	12 „ „

the balance being incombustible.

For the carbon we have

$$\begin{aligned} & 0.86 \times \frac{32}{12} \times \frac{100}{23} \\ &= 0.86 \times 11.6 \\ &= 9.98 \text{ pounds of air.} \end{aligned}$$

For the hydrogen we have, similarly,

$$\begin{aligned} & 0.12 \times 8 \times \frac{100}{23} \\ &= 0.12 \times 34.8 \\ &= 4.176 \text{ pounds of air,} \end{aligned}$$

giving a total of $4.176 + 9.98 = 14.156$ pounds of air per pound of fuel ; or a fuel to products ratio of

$$\frac{1}{15.156} = 0.066.$$

Now the heating value of this fuel per pound is equal to

$$(0.86 \times 14540 + 62000 \times 0.12) 0.95,$$

and per

$$0.066 \text{ pound, } 1250 \text{ B.Th.U.,}$$

allowing 5 per cent. for the heat of formation of the fuel oil compounds.

If, now, this heat were applied to unit mass of the products, we should have

$$Q = \int_{T_1}^{T_2} \frac{C'_p dT}{m_m},$$

where m_m is the apparent molecular weight of the mixture.

Substituting for C'_p , $a_p + bT$, and working out the numerical value of m_m , we have

$$\frac{5 \times 1250}{9} = \int_{978}^{T_2} \frac{(6.692 + 0.001T)dT}{29.04},$$

which gives

$$\begin{aligned}
 695 &= \int_{978}^{T_2} (0.231 + 0.0000344T) dT \\
 &= 0.231 (T_2 - 978) + \frac{0.0000344(T_2^2 - 978^2)}{2}
 \end{aligned}$$

whence T_2 is equal to 3261 deg. Cent. abs. or 5870 deg. Fahr. abs.

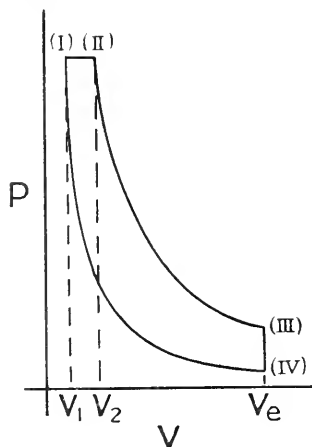


FIG. 51.—Pressure-Volume Diagram for Diesel Engine.

Such temperatures, however, are not realized, due to the presence of surplus air and heat suppression, that is, heat lost to the walls and through dissociation, etc.

Maximum Cut-off with Complete Oxygen Utilization.—Let us consider the ideal PV diagram for the Diesel engine with a view to determining the position of the point of cut-off when this is as late as it could be ideally, that is, with no air of dilution present.

Let the compression ratio be 35, as above. Let also the initial temperature be 200 deg. Fahr. Then

$$P_1 V_1^\gamma = P_4 V_4^\gamma$$

and
$$\frac{V_2}{V_1} = \frac{T_2}{T_1}$$

whence
$$\begin{aligned}
 \frac{V_2}{V_4} &= \frac{T_2}{T_1} \left(\frac{P_4}{P_1} \right)^{\frac{1}{\gamma}} \\
 &= \frac{3261}{978} \left(\frac{1}{35} \right)^{\frac{1}{1.38}} \\
 &= 25.34 \text{ per cent.}
 \end{aligned}$$

With lower initial temperature of the charge, the cut-off would be found to be somewhat later, reaching a

maximum value of about 33 per cent. when the temperature of the induction air is 60 deg. Fahr. The combined effects of excess air and heat suppression will produce proportionate changes in the ratio of V_2 to V_4 , this quantity becoming less as these effects increase.

The Indicated Mean Pressure.—The mean pressure of the indicator card, resulting from any given position of the cut-off, may be investigated from the equation

$$P_m = \frac{P_1(V_2 - V_1) + \frac{P_2V_2 - P_3V_3}{\gamma - 1} - \frac{P_1V_1 - P_4V_4}{\gamma - 1}}{V_4 - V_1}$$

Taking the case already cited, the initial pressure being 14.3 pounds per square inch, and temperature 200 deg. Fahr., the mean pressure with maximum oxygen utilization, and without any heat suppression, works out at 181 pounds per square inch.

From tests made by Professor Mellanby of the Royal Technical College, Glasgow, on a four-cycle air-injection engine, it is concluded that the curve relating indicated mean pressure and oxygen content in the exhaust suggests 140 pounds per square inch as a limiting value. The present working limit, without supercharging of the cylinder, is about 125 pounds per square inch, with an average value around 100 pounds per square inch.

The actual difficulty, however, is not to secure higher indicated mean pressures by reducing the proportion of excess air, but to prevent injury to the crown of the piston and to the walls of the cylinder, as the temperature of the cylinder contents rises with increase in the proportion of fuel burnt per stroke.

Just below the skin of the piston face and cylinder walls the periodic temperature fluctuations are damped, and in the main body of the metal there is a steady temperature gradient. When, however, the amount of heat to be transmitted per unit area of cooling surface increases, either by reason of increase in cylinder diameter, increase of mean pressure (and therefore of maximum temperature) or increase of speed, there comes a point where injury to

the metal surfaces exposed to those high temperatures results, and a series of fine cracks make their appearance.

Hence, for a given mean pressure and speed there is a limiting diameter of cylinder, which it is at present impossible to increase. The ideal will be reached when the limiting diameter of piston will be independent of the rate of heat generation.

The practical difficulties are aggravated by the severe conditions under which combustion takes place, the high oxygen density increasing the tendency to injure the walls of the cylinder and burn the piston crown.

As an example of the way in which these difficulties have been met, and as showing the trend of evolution of the marine Diesel in general, and certain novel and interesting features in particular, there follows a description of the engines of the motor ship *Swanley*, built by Messrs. Barclay, Curle and Co., Ltd., and engined by the North British Diesel Engine Company, Ltd., Glasgow.

THE NORTH BRITISH SLIDING CYLINDER DOUBLE - ACTING TWO - CYCLE DIESEL ENGINE

Introduction.—As already indicated, Diesel engines are broadly classified under the terms ‘four-stroke cycle’ and ‘two-stroke cycle.’ There is a further differentiation into single-acting and double-acting engines, meaning that in the former case the pressure of the charge acts on one end of the piston only as in the typical automobile engine, while in the latter both ends of the cylinder are put to use. While a large development of the four-cycle single-acting engine for marine purposes has taken place, it appears for several reasons as though the main line of evolution must proceed by way of the two-cycle double-acting type. Not only are four-cycle single-acting engines possessed of greater bulk and weight, and hence have a higher first cost, but in order to be directly reversible they must have at least six cylinders, and for various

reasons should not have more than eight. Hence, standardization of one type having a constant cylinder diameter, but a variable number of cylinder units for a large range of powers, becomes impossible. On the other hand, the two-cycle double-acting type may be made directly reversible with any number of cranks from two to eight (just as in the case of the compound or multiple expansion steam-engine). This enables any one size of unit to be adopted for a very wide range of powers, thus facilitating standardization and mass production to an extent much greater than has previously been possible with any type of prime-mover. For example, a double-acting two-cycle cylinder developing 450 B.H.P. may be used in directly reversible engines of from 900 (i.e. 2×450) for a single screw, up to 7200 (i.e. two banks of eight cylinders) for twin screws.

CONSIDERATIONS GOVERNING THE ADOPTION OF THE FEATURES PECULIAR TO THE NORTH BRITISH DIESEL ENGINE

Elimination of the Piston Rod.—Practically all previous attempts to solve the problem of the two-cycle double-acting engine have followed standard steam-engine design in that they have been of the piston rod type. In a steam-engine the piston rod does not present any difficulties, but the working conditions in the Diesel engine are very different, and while the piston rod has not proved very troublesome or dangerous in use, it will doubtless be generally admitted that the presence of a cool rod in the heart of the combustion space cannot be conducive to more efficient combustion. There is further an interference with effective scavenging and an added complication of the lower cylinder cover, together with the necessity for a high-pressure stuffing box. It would appear from every point of view that the elimination of the piston rod is desirable if this can be accomplished without undue mechanical complication. It is further

likely that the problem of efficient combustion, with which is bound up the proper form of combustion space, will be of even greater importance in the future than it is at present.

The larger sizes of marine Diesel engine will doubtless be called upon to operate with some cheap quality of oil, probably boiler fuel oil, which contains usually a large percentage of asphaltum (the hard bituminous residuum of crude oil).

Consequently the difficulties of securing efficient combustion will be increased.

The Sliding Cylinder.—In order that the temperatures may be kept down, and that the cylinder may have a charge of pure, comparatively cool air, scavenging must be efficient.

In order, therefore, to eliminate as far as may be possible any uncertainty in this regard, the cylinder of this engine is made to reciprocate synchronously with the piston, uncovering at the appropriate time a row of scavenging ports around the head end of the cylinder and causing a blast of air from the reservoir of the pump to sweep the cylinder from end to end.

Principles and Construction of the North British Diesel Engine.—The design is illustrated by figs. 51, 52 and 53, which are partly diagrammatic. Fig. 52 is an athwartship section. So also is fig. 53, which shows the exterior of one unit, and fig. 54 is a longitudinal section through one unit.

Referring to these illustrations, a unit of the engine consists of an upper cylinder A and a lower cylinder A₁, both of which are open at each end. The cylinders, together with their water jackets B and B₁ and scavenging and exhaust branches D and D₁, respectively, are free to reciprocate relatively to the two cylinder covers E and E₁ provided at the outer ends of the cylinders, the covers being made in the form of stationary pistons and furnished with spring rings to ensure gas tightness. The upper and lower cylinders are connected by two ties, F and F₁, diametrically opposed to one another. A double-ended piston G, with gudgeon pin H, reciprocates inside

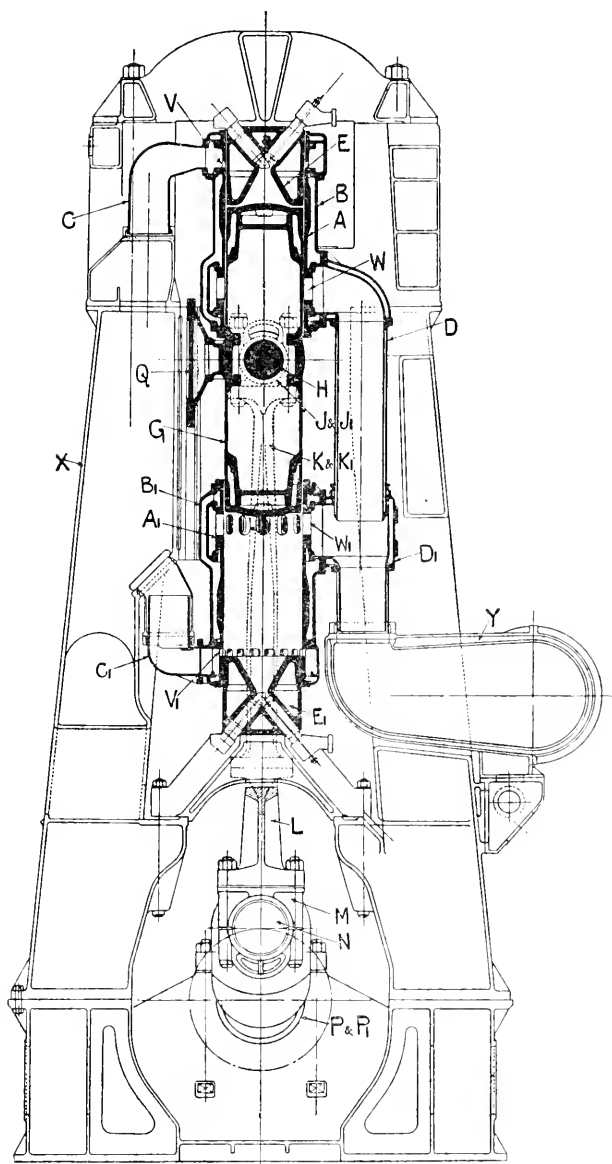


FIG. 52.—Athwartship Section of North British Diesel Engine.

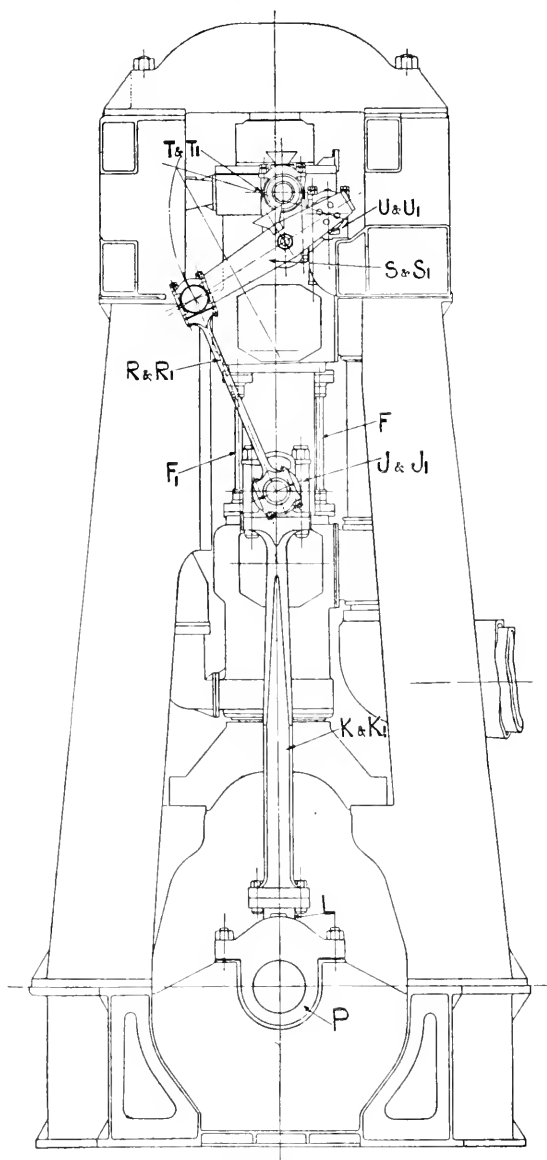


FIG. 53.—Athwartship Elevation of North British Diesel Engine.

the cylinders and between the covers E and E_1 . The projecting ends of the gudgeon pin pass out between the upper and lower cylinders, the space between the cylinders being sufficient to accommodate their movement. Journals are provided towards the ends of the gudgeon pin to take the top end bearings J and J_1 of the connecting-rod fork L . At the lower end of the connecting-rod fork a crank-pin bearing M is provided, which operates on the crank N . Crankshaft bearings P and P_1 of the usual form are provided on each side of the crank N . A guide Q is directly attached to the middle of the piston. The cylinders, together with their water jackets and scavenging and exhaust branches, are caused to reciprocate by the link gear R, S, T, U (figs. 53 and 54). The links R and R_1 are attached at their lower ends to extensions of the main gudgeon pin, and at their upper ends to journals provided at the extreme ends of the levers S and S_1 . The cylinders are operated by means of links T and T_1 , which are attached at intermediate points on levers S and S_1 . The levers swing about fulcrum bearings U and U_1 . Therefore the cylinders A and A_1 , together with their water jackets,

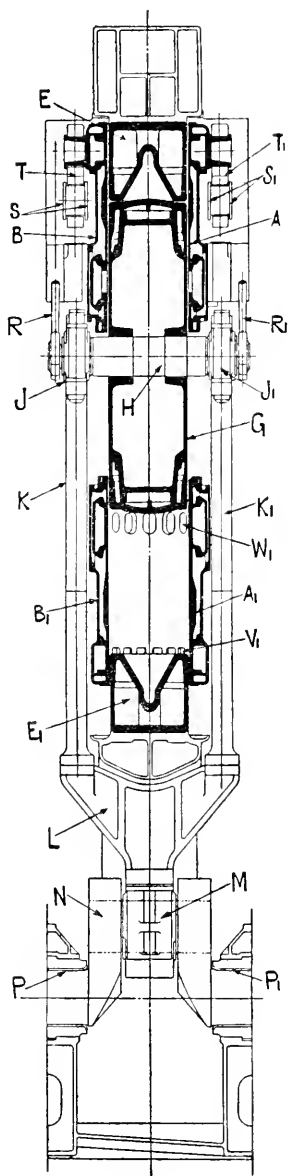


FIG. 54.—Longitudinal Section through one unit of North British Diesel Engine.

reciprocate synchronously with the main piston, but through a much smaller distance. Thus the scavenging ports V and V_1 provided at the outer ends of the cylinders are uncovered by the relative movements of the cylinders and covers E and E_1 , and the exhaust ports W and W_1 provided at the inner ends of the cylinders are uncovered by the relative movement of the piston to the cylinders. The front column X is used as a scavenging air reservoir, and suitable stuffing boxes are provided on it for the sliding scavenging branches C and C_1 .

The exhaust manifold Y has a stuffing box to take the exhaust sliding branch D .

The first large engine of this type was built in 1922 for the motor ship *Swanley*. It consisted of three units, each of $24\frac{1}{2}$ -inch bore by 44-inch stroke, designed to give 2000 B.H.P. at 100 r.p.m. The figures show the engine in elevation and plan. An air compressor and scavenging pump are provided at the forward and aft ends respectively.

Under test, the best consumption over an extended run was from 0.448 to 0.45 pound of boiler fuel per B.H.P. hour, with a mechanical efficiency in the neighbourhood of 75 per cent. No important troubles developed during the maiden voyage of the ship, but a few minor alterations were made in the design of the main engines after her return.

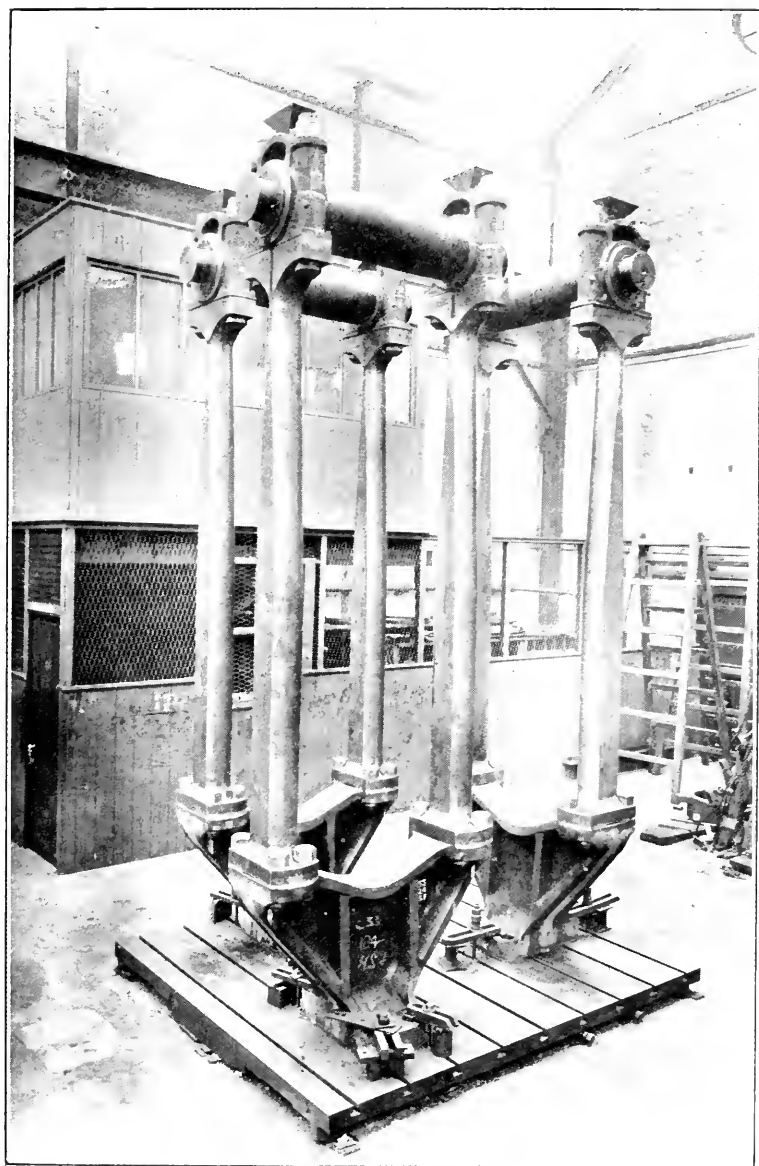


FIG. 55. Forked Connecting Rods of North British Diesel Engine.

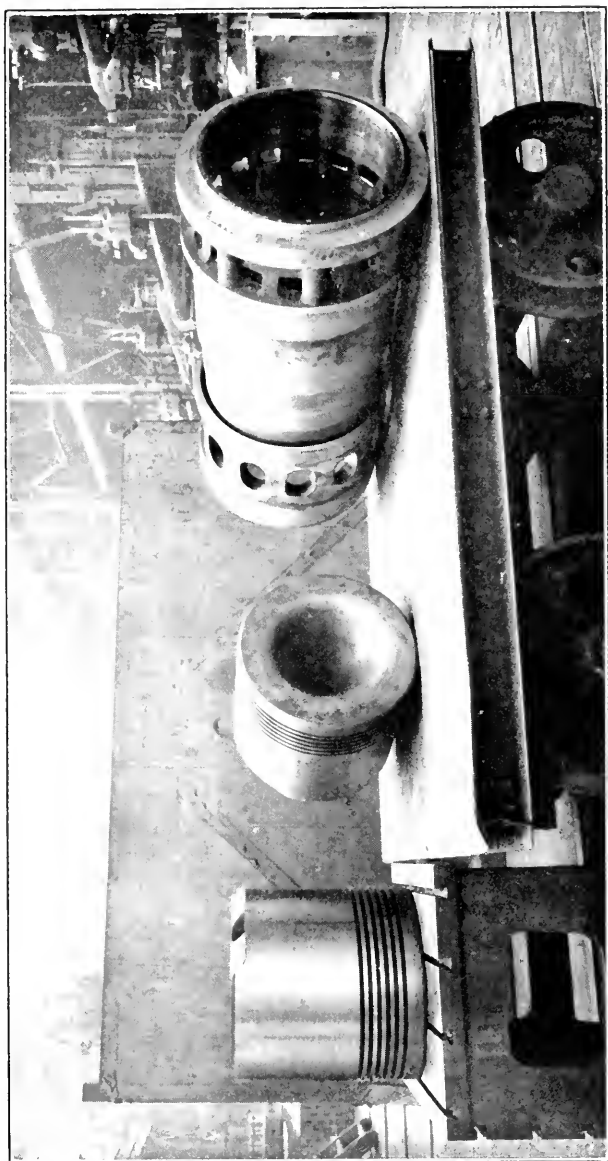


FIG. 56. Cylinder Heads and Liner of North British Diesel Engine.

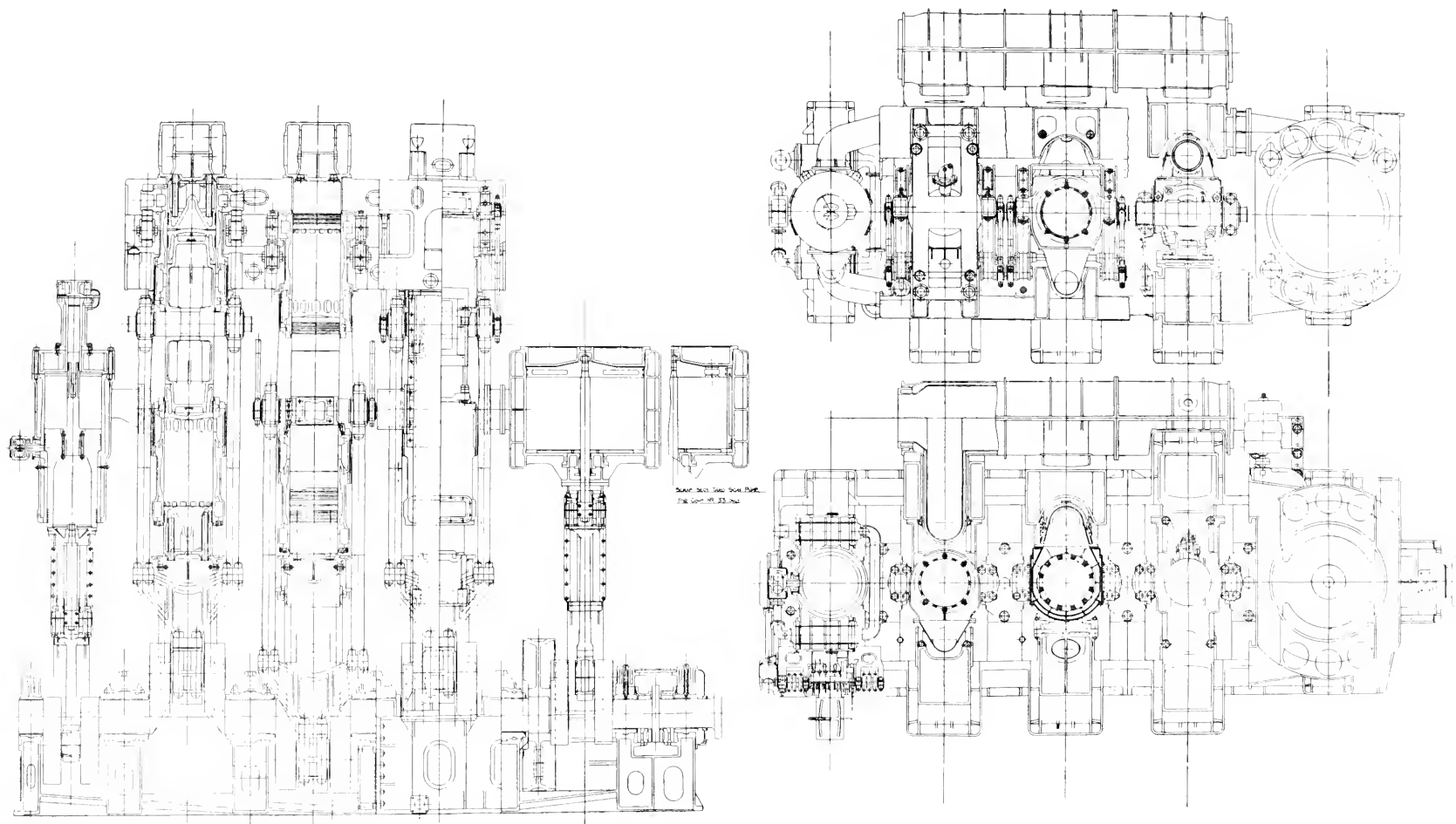


FIG. 57. ENGINE OF THE M.S. "SWANLEY" IN ELEVATION AND PLAN.
PLATE X.



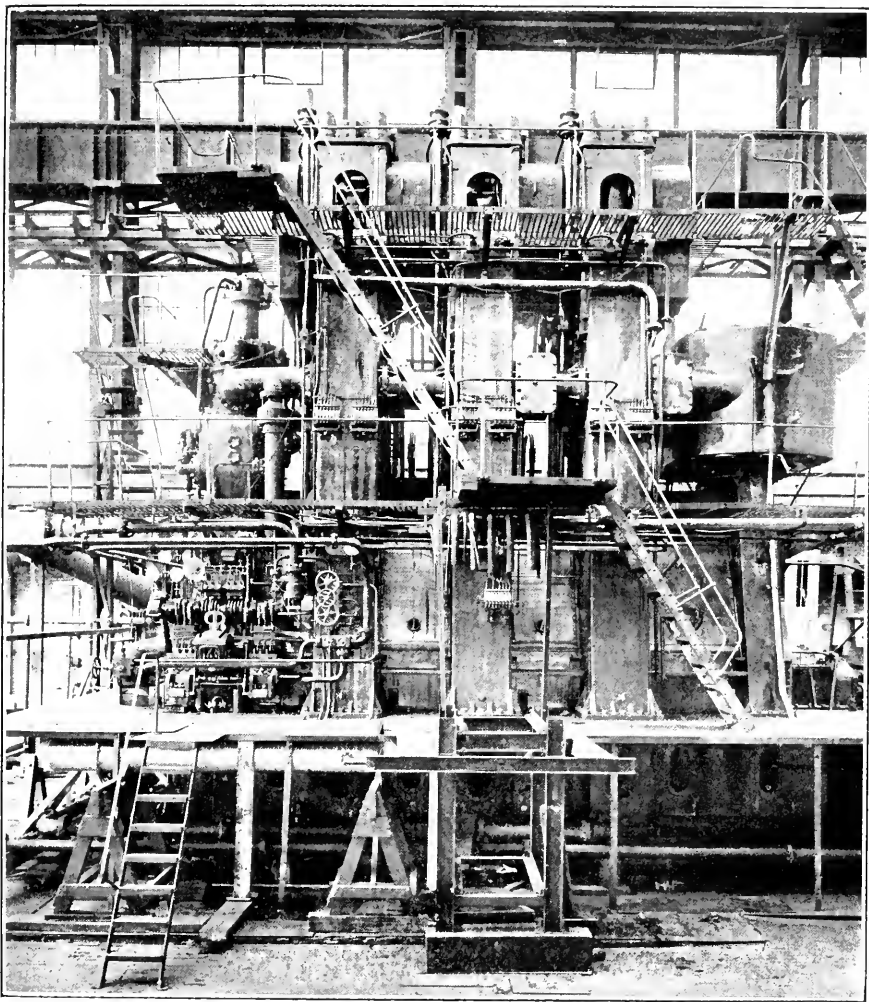


FIG. 58. Engine of the M.S. "Swanley,"
built by North British Diesel Engine Co. Glasgow.

THE GAS-TURBINE¹

Introduction.—Inquiries are frequently made concerning the possibility of developing the internal combustion turbine on a scale comparable with that of the steam-turbine. The reason for the failure of the gas-turbine to compete successfully with other major prime-movers lies in the difficulties inherent in the problem itself rather than in any lack of enterprise and complete consideration of its possibilities. Many attempts have been made to design a machine which would compare favourably, in point of efficiency, reliability and cost, with the steam-turbine, but failure has attended most of these efforts. The purpose of the present discussion is to give some indication of the overall efficiencies which may be expected when the limitations imposed by the comparatively high temperatures to which the working parts are exposed are taken into consideration. Of these latter, the blades and rotor alone are of necessity exposed to high stresses, and since no water-jacketing is possible, the temperature of the casing within which the disc rotates must be maintained below a certain limit, dependent upon the properties of such materials of construction as are available. This ‘casing temperature,’ as it is termed, will, in general, be higher than the temperature of the gases discharged from the nozzles on account of friction and eddy loss during the passage of the gas across the vanes, in windage losses, i.e. disc friction, and due to the transformation of the residual kinetic energy, all of which losses become heat.

Professor Marks states that the present practicable

¹ The author is indebted for his information concerning the gas-turbine to an original paper, ‘Gas-Turbines,’ by Professor Marks, published in the *Trans. of the Am. Soc. of Mech. Eng.*, 1924.

limit of casing temperature appears to be about 850 deg. Fahr., and mentions that the Holzwarth turbine operates at this temperature with vanes of soft carbon steel. Professor Rateau, however, reports a temperature of 1200 deg. Fahr. in his exhaust gas-turbine with rotor of tungsten tool steel. Reference will be made to this turbine later. As, however, most steels lose about half their tensile strength around 1000 deg. Fahr., progress will be arrested until the metallurgical problems involved in the production of a material which will retain a greater proportion of its tensile strength at high temperature are solved. While the problem of high temperature in its relation to existing materials of construction is a formidable one, this is unfortunately not the only issue involved. In order to secure the advantages of large power output for a given size and weight, it is necessary to employ a compressor of the centrifugal type for the compression of the induction air. Unless such an arrangement is adopted there is little point in constructing a turbine at all. If a compressor of the reciprocating type be used, no advantage in this regard is obtained over the ordinary reciprocating internal combustion engine. Until, therefore, improvement is effected in centrifugal compressor design, comparatively large losses may be expected from this cause.

It may therefore be inferred from the foregoing that whereas by comparison the steam-turbine, being able to use a more extended pressure and therefore temperature range, is more favourably situated than the ordinary counter-flow steam-engine, the gas-turbine possesses no such advantage over the reciprocating internal combustion engine; for, while a certain gain is to be expected from complete expansion, the limitation imposed on combustion temperatures more than offsets this advantage. There is, however, a possibility that the decrease in weight and first cost due to the absence of boilers and condenser might incline opinion in its favour, even though the brake thermal efficiency is not likely to exceed that of a first-class steam-turbine plant and the fuel is likely to be more costly. A review of the possibilities of the gas-turbine

does not, however, give much hope of the realization of efficiencies such as would encourage attempts to overcome the many difficulties with which the machine is surrounded.

It may nevertheless be of interest to examine possible cycles of the gas-turbine in order to form some estimate of what these efficiencies may be.

Cycle of the Gas-Turbine.—This may be considered most readily by reference to the temperature-entropy chart.

Consider again the expressions already given for the change of entropy of a gas or gaseous mixture, the variation of the specific heat being taken account of.

$$\phi - \phi_o = a_v \log_e \frac{T}{T_o} + b (T - T_o) + \frac{R'}{J} \log_e \frac{V}{V_o}$$

$$\text{and } \phi - \phi_o = a_p \log_e \frac{T}{T_o} + b (T - T_o) - \frac{R'}{J} \log_e \frac{P}{P_o}.$$

Since a_v and a_p are constant for all gases, but b may vary, it is possible to construct a chart which will answer for all gases (per, of course, a mass in each case numerically equal to the molecular weight) by making use of the following device.

Let, for the sake of convenience, the axis of zero entropy be drawn vertical for a value of $b = 0.001$. Then, for any gas for which the value of b differs from this figure, let a new axis of zero entropy be set up but inclined to the former at such an angle that entropy differences due to the change in b are measured by the horizontal intercept between these axes.

Referring to fig. 59, OA is the axis of zero entropy for gases having the particular value 0.001, OB is the axis of zero entropy for gases having some other (greater) value of b . In all cases adiabatics for the gas in question lie parallel to the appropriate axis.

Then the cycle is traced as follows. The fresh charge is compressed along Oi in the case of adiabatic compression, where Oi is drawn parallel to OA connecting the point of zero entropy with the value of b for the fresh

waste gases are applied to the purposes of regeneration or to the raising of steam in a boiler, their condition on leaving is indicated by the point 5.

The Determination of Efficiencies of the Gas-Turbine.—The effect of combustion upon the fresh charge is to change the number of molecules. Since, of course, the mass remains constant, it follows that if

m be the apparent molecular weight of the mixture before combustion, and if

m_1 be the apparent molecular weight of the mixture after combustion, and also if

n be the number of mols. present before combustion, and

n_1 be the number of mols. present after combustion, then

w the weight of the working substance $= mn = m_1 n_1$.

(Since the mol. is, by definition, a mass of the substance numerically equal to the measure of the molecular weight,

$$\frac{w}{m} = n.)$$

For all constant-pressure cycles the available total heat drop is given by

$$W_{tp} = I_2 - I_3$$

and for constant-volume combustion

$$W_{tr} = I_{2'} - I_3 - V_{2'} (P_{2'} - P_1) 1/J$$

The total heat may be determined from

$$I = \int_{T_o}^T C'_p dT = a_p (T - T_o) + \frac{b (T^2 - T_o^2)}{2}$$

and the internal energy from

$$E = \int_{T_o}^T C'_v dT = a_v (T - T_o) + \frac{b (T^2 - T_o^2)}{2}$$

For the work done during adiabatic compression, we have

$$W_{ca} = I_1 - I_0$$

and for the work done during isothermal compression,

$$W_{ci} = \frac{R}{J} T_2 \log_e \frac{P_1}{P_2},$$

in which T_2 is the constant temperature during the isothermal compression of the fresh charge, and P_1 and P_2 are the upper and lower limits of pressure, respectively, between which the compressor works.

For the mass of the working substance which takes part in the action, the work delivered is

$$n_1 W_t E_t$$

in which, as above,

n_1 is the number of mols. present after combustion,
 W_t is a measure of the work ideally available, and
 E_t is the efficiency of the turbine.

Similarly, the work which must be done upon the compressor is

$$n \frac{W_c}{E_c}$$

in which, as above,

n is the number of mols. present before combustion,
 W_c is a measure of the work ideally necessary for compression, and
 E_c is the efficiency of the compressor.

The net work actually available is therefore

$$W = \left(n_1 W_t E_t - n \frac{W_c}{E_c} \right)$$

Possible Gain in Efficiency from the more Complete Expansion in the Gas-Turbine.—When this question is investigated without regard to the limitation of the casing temperature, and neglecting all friction and other losses, it is found that the gain in cycle efficiency is greatest with low ratios of compression. With, for example, a ratio of compression of 4 we find an increase in the cycle efficiency of from 27·8 to 39·3 per cent., or

an increase of 41·4 per cent., the assumed conditions being as follows :

Initial pressure, 14·7 pounds per square inch.

Initial temperature of the charge, 200 deg. Fahr.

Final pressure, 14·7 pounds per square inch, and isothermal compression of the charge.

Overall Efficiency of the Turbine.—On account of the necessity for limiting the casing temperature of the gas-turbine, the excess air drawn in with the fuel must be increased very largely.

This has the unfortunate result that the ratio of positive work done by the turbine, W_t , to the negative work of precompressing the air before combustion, W_c , is much less than in the ordinary reciprocating engine. This ratio, which we may call R , is of considerable importance in determining the efficiency of the gas-turbine. The compression work has to be done in a separate machine, which would presumably always be a centrifugal compressor for the reason already mentioned. The net work of the turbine is then

$$W = W_t E_t - \frac{W_c}{E_c},$$

where E_t and E_c are the efficiencies of the turbine and compressor respectively. This may also be written

$$W = W_t \left(E_t - \frac{1}{RE_c} \right).$$

In actual practice the value of E_t would not be greater than 0·7, and the efficiency of a centrifugal compressor E_c for high ratios of compression on the isothermal basis is probably less than 0·7. Examining the factor

$$\left(E_t - \frac{1}{RE_c} \right)$$

it will be seen that each of the three quantities should be as large as possible. With E_t and E_c both equal to 0·7 this factor becomes negative, for R equals 2. This means that no work is available from the turbine under these circumstances.

Similarly, for E_t and E_c both equal to 0.6, R must be greater than 2.77 in order that the turbine may be able to operate at all.

Possibility of Utilizing the Heat in the Exhaust Gases.—In any gas-turbine it will be found that a considerable fraction of the heat of combustion escapes with the exhaust gases. A good turbine performance is not possible unless this heat is utilized.

Various procedures have been used or seem of sufficient promise to justify analysis.

1. The exhaust gases may pass through a regenerator for heating the fresh air. With a counterflow arrangement the entering air may be heated, after compression, to the temperature of the exhaust gases.

2. The exhaust gases may generate steam which

- (a) May be sent into the combustion space in a constant pressure turbine, or
- (b) May expand through steam nozzles and do work on the buckets (thereby cooling them), or on a special turbine on the same shaft, or
- (c) May drive a special condensing steam turbine and supply part or all of the power for operating the compressor ; or
- (d) Various combinations of the above may be used.

Pressure Limits in Gas-Turbines.—There is no special condition limiting the compression pressure used in a gas-turbine, but the increase in efficiency from high compression pressures is less than in reciprocating engines because the efficiency of centrifugal compressors falls off at high pressures in consequence of the high density of the air in the later stages and the resulting increase in frictional losses. The exhaust pressure will be in excess of atmospheric pressure when the heat of the exhaust gases is utilized in a regenerator or boiler, unless a gas exhauster is employed. As an exhauster acts on the gases after their volume has been diminished by cooling, its use may show an increase in thermal efficiency.

Many modifications of the gas-turbine other than those mentioned above have been suggested and employed.

The injection of water into the combustion space in conjunction with preheating the compressed air by the exhaust gases is analysed below. There have been several so-called gas-turbine projects employing an oscillating water column (as in the Humphrey pump) and a hydraulic turbine. These are really water-piston engines and are not considered here.

Explosion Turbines.—Gas-turbines may be classified in the same two groups as reciprocating engines, constant-volume-combustion or explosion turbines, and constant-pressure-combustion turbines. These are strongly differentiated.

In the constant-volume-combustion or explosion turbine the action is intermittent. The compressed air and fuel are introduced into the combustion chamber, the admission valve is closed, and the mixture is ignited and exploded; the nozzle valve is then opened and the products of explosion pass through the nozzle to the turbine buckets. After the pressure in the combustion chamber has fallen sufficiently, cooling air may be sent through the combustion-chamber nozzle and on to the turbine buckets, as in the Holzwarth turbine. The nozzle valve is then closed and the cycle begins again. Attention may be drawn here to certain features of this process. In order to operate, it is essential that the mixture in the combustion chamber should be explosive at the moment of ignition. This requirement limits the permissible amount of excess air, but to an extent which is not determinable from existing data on explosive mixtures. With air preheating, the temperature of the charge will be high—a condition favourable to ignition. If the fuel is injected very rapidly into air preheated to the ignition temperature and with the great turbulence which would exist under those conditions, the combustion may take place, as in a Diesel engine, with any quantity of fuel, however small; but the combustion, though very rapid, will not be a true explosion.

Another special condition which is inherent in the explosion turbine is that the spouting velocity of the gases during the expansion period will fall from a maximum

at the instant of opening the nozzle valve to zero when the pressure in the combustion chamber is equal to that in the turbine casing. This is not a condition favourable to high turbine efficiency. Stodola shows for a particular case, with initial spouting velocity of 4526 feet per second, that the velocity will have fallen to 3280 feet per second when 66 per cent. of the charge has passed, and to 1640 feet per second after 96 per cent. has passed. He finds a mean turbine efficiency of 63.5 per cent. in this case as compared with an efficiency of 70 per cent. for the most favourable gas velocity. These turbine efficiencies do not take windage losses into account.

Constant-Pressure-Combustion Turbine.—In the constant-pressure-combustion turbine there is a steady flow of air and fuel to the combustion chamber and a steady discharge of products of combustion through the nozzles. The temperature conditions offer much more difficulty than with the explosion type. Temperatures must be kept down either by using a very large excess of air, by injecting water or steam into the combustion space, by separate air or steam jets acting on the buckets, or by other device. This type, which has given highest efficiencies in reciprocating (Diesel) engines, does not promise similar efficiencies in the gas-turbine in consequence of the low ratio of positive to negative work, R , and the low value of compressor efficiency, E_c , which results from the high compression pressures. On the other hand, turbine efficiencies, E_t , are higher than in the explosion turbine, as indicated in the preceding paragraph.

Fuels.—Either gaseous or liquid fuels may be used in a gas-turbine. Gaseous fuels must be precompressed and may add seriously to the negative work of the cycle, especially if the gases are of low heating value and the excess air is low. Liquid fuels require injection devices similar to those used in Diesel or semi-Diesel engines. The form of the combustion space can be made more favourable than is possible in reciprocating engines.

Conditions for High Efficiency.—The conditions assumed for the calculations are in some respects not

practicable at present—this is especially true as regards casing temperatures, as already noted. The efficient gas-turbine must wait for metallurgical developments which will yield a metal better able to retain high strength at high temperatures than any now available.

No attempt is made to calculate probable turbine efficiencies, E_t , but it is evident that certain conditions, such as the reduction of windage losses, will favour higher efficiencies. The constant-pressure-combustion turbine, exerting a constant torque, will have a smaller percentage windage loss than the explosion turbine with intermittent applications of a lower average torque. A turbine with back pressure reduced by an exhauster will also have a smaller windage loss. On the other hand, an explosion turbine with short expansion period followed by prolonged cooling by low-pressure air, will have a low turbine efficiency if the conditions are such that the turbine acts like a blower and speeds up the cooling air as it passes through the buckets. Stodola estimates the use of over 16 per cent. of the turbine power to carry out the cooling by low-pressure air in this manner. With high-pressure air it is probably not less.

Efficiencies of Various Cycles.—The calculation of the efficiency of a gas-turbine, using variable specific heats, can be carried out most readily by the use of a temperature-entropy chart such as is given by Stodola. The process of calculation and the values of the specific heats have already been given. In most of the cases considered these calculations have been made for some assumed casing temperature and for a series of compressor and turbine efficiencies. The turbine efficiencies cover all the losses, including nozzle friction, bucket friction, windage, residual kinetic energy, and machine losses; that is, they are the ratio of brake work to the adiabatic total-heat drop from combustion-chamber conditions to exhaust pressure. All the energy lost in the turbine is assumed to be used in heating the gases at constant pressure from the temperature at the end of adiabatic expansion to the casing temperature. With a fixed casing temperature and a fixed combustion pressure this

requires a different combustion temperature for each assumed turbine efficiency.

Certain conditions have been assumed for all the cases considered—with a few exceptions as noted :

- (a) When gaseous fuel is used, it is assumed to have the volumetric composition : $H_2 = 11$; $CO = 22$; $CO_2 = 10$; $N_2 = 57$ per cent., and a low heating value of 102 B.Th.U. per cubic foot at 14.7 pounds per square inch abs. and 60 deg. Fahr.
- (b) When oil fuel is used, it is assumed to have the weight composition : $C = 86.5$; $H_2 = 11.2$; $N_2 = 2.3$ per cent., and a low heating value of 18,610 B.Th.U. per pound.
- (c) The effect on volume of molecular shrinkage due to combustion is neglected ; the resulting error in the calculated efficiency is less than 1 per cent. The change in chemical composition is in other respects taken into account, as already shown.
- (d) The heat of combustion is assumed to be the same at constant pressure and at constant volume. The actual difference is less than 0.5 per cent. Similarly, the variation of heat of combustion with temperature is neglected.
- (e) The pressure drop between the air compressor and the combustion chamber is neglected.
- (f) Compression efficiencies are all on the isothermal basis. The compressed air is supposed to have the room temperature when it reaches the regenerator or combustion chamber.
- (g) The radiation loss to the walls of the combustion chamber is assumed to be 10 per cent. of the heat of combustion. In most cases this heat is regarded as lost, but in certain specified cases it is utilized in generating steam.
- (h) The pressure of the exhaust gases before passing through the regenerator or steam boiler is generally assumed to be 17.06 pounds per square inch abs. when exhausting to the atmosphere.

- (i) Most of the calculations assume an initial air pressure of 14.22 pounds per square inch abs., which is 3 per cent. less than the standard atmosphere. The difference between this pressure and the standard atmosphere may be considered as the pressure drop of the air entering the compressor. Its effect on efficiency is quite negligible.
- (j) Spouting velocities are always theoretical quantities assuming no nozzle loss.

CALCULATED PERFORMANCE OF GAS-TURBINES

(a) **Explosion Turbine with Regeneration.**—The highest turbine-casing temperature which seems likely to be practicable, according to present indications, is 932 deg. Fahr. With exhaust gases at that temperature, and with an efficient regenerator, a temperature of 842 deg. Fahr. for the charge entering the combustion chamber should be obtainable. The efficiencies attainable under these conditions and with compression-pressure ratios 5, 10, 15 and 20 are given in Table I. (p. 169). Blank spaces indicate negative values of the brake thermal efficiency, i.e. the turbine work is not sufficient to drive the compressor. The temperatures after adiabatic expansion are the temperatures at which the gases would leave a frictionless expansion nozzle. The reheating due to nozzle and bucket friction, windage losses, and residual kinetic energy is the difference between the casing temperature and the temperatures after adiabatic expansion; in this case it ranges from 146 to 470 deg. Fahr. The amount of excess air is so great (ranging from 533 to 1796 per cent.) that no explosive mixture is possible; the cycle would presumably have to be carried out by injecting the fuel rapidly as in a semi-Diesel engine. The spouting velocities are the maximum values (disregarding nozzle friction) and diminish from the given values to zero. They are moderate and can be readily taken care of without necessitating excessive peripheral speed for the turbine

buckets. The ratio of bucket speed to spouting velocity should be lower for gas-turbines than for steam-turbines because of their greater windage loss.

(b) Explosion Turbine with Regeneration and Cooling.—The conditions of Table I. have been selected as apparently those giving maximum efficiency for an explosion gas-turbine with regenerator but without any special cooling device. If the gases reach a temperature in excess of 932 deg. Fahr. in the turbine casing, it becomes necessary to add some device to keep down the temperature of the buckets and turbine disc. If the cooling device is a jet of high-pressure air expanding on to the buckets, it will consume power for compressing the air, and all of this power will not be recovered at the buckets. If it is low-pressure air the buckets will do work on it. The magnitude of the power required for this purpose depends upon the details of the particular scheme used.

Stodola gives charts showing the efficiencies of explosion turbines with regenerative heating of the compressed charge to 1022 deg. Fahr., and with casing temperatures of 1112, 1472 and 1832 deg. Fahr. Table II. (p. 170) has been compiled from these charts. The tabulated values cannot be regarded as attainable unless there should be developed turbine-rotor materials capable of withstanding these high casing temperatures. The additional work required for cooling the rotor may be taken into account by assuming lower values of the compressor or turbine efficiency. With prolonged cooling by low-pressure air, alternating with the discharge of the hot explosion gases, the windage loss of the turbine will be much increased and the turbine efficiency correspondingly reduced. The higher the casing temperature (calculated on the assumption of no air cooling) the greater will be the negative cooling work.

(c) Explosion Turbine with Regeneration and Water Injection.—Another possible method of lowering the casing temperature is by injecting water into the combustion space. This case has been calculated for combustion with 100 per cent. excess air, a casing temperature of 932 deg. Fahr., regenerative heating of the compressed

TABLE I.—CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITH REGENERATIVE HEATING OF THE COMPRESSED CHARGE TO 842 DEG. FAHR., AND TURBINE-CASING TEMPERATURE 932 DEG. FAHR.

Assumptions : Back pressure at gas-turbine, 17·06 lb. per sq. in. abs.; gas fuel; heat loss to walls of combustion chamber, 10 per cent. of heat of combustion; compression efficiencies on isothermal basis.

Compression pressure, lb. per sq. in. abs.																

Blank spaces under * Brake thermal efficiency indicate negative values, i.e. the turbine work is not sufficient to operate the compressor.

TABLE II.—BRAKE THERMAL EFFICIENCIES, PER CENT., OF EXPLOSION GAS TURBINES WITH REGENERATOR

Data from Figs. 1048, 1049 and 1050 of Stodola's *Dampf- und Gas-Turbinen* (5th edition).

Assumptions: Back pressure at gas-turbine, 17.06 lb. per sq. in. abs.; fuel, producer gas with low heating value (123 B.Th.U. per cu. ft.) at 1 atm. pressure and 60 deg. Fahr.; regenerative heating of compressed air to 1022 deg. Fahr.; heat loss to combustion chamber, 10 per cent. of heat of combustion; compression efficiencies on isothermal basis.

Compressor efficiency, } per cent.		Turbine efficiency, per cent.																								
		55					60					65					70					75				
		50	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75
Casing temperature, 1112 deg. Fahr.																										
71.3	—19.0	—12.7	—6.0	0.0	4.2	—10.3	—3.6	2.2	7.5	11.0	—2.7	3.7	9.4	14.0	17.2	4.1	10.1	15.2	19.4	22.8	10.4	15.0	20.5	24.5	28.0	
142.2	—14.5	—7.0	—1.0	4.9	9.1	—4.7	2.1	7.9	13.2	16.2	4.1	10.0	15.4	20.3	23.7	11.5	17.0	22.0	26.3	29.9	18.8	24.8	28.5	32.3	36.1	
213.0	—13.7	—6.0	0.5	6.0	10.5	—3.6	3.4	9.0	14.4	18.2	5.5	11.6	17.0	21.9	25.2	13.5	19.0	23.7	28.2	31.7	20.8	26.0	30.5	34.2	37.5	
285.0	—14.0	—6.3	0.5	6.0	10.5	—3.6	3.4	9.2	14.5	18.7	5.8	12.0	17.3	22.3	26.0	14.3	19.4	24.5	29.0	32.8	21.5	26.8	31.0	35.0	38.0	
Casing temperature, 1472 deg. Fahr.																										
71.3	0.4	3.7	6.7	9.2	11.3	5.0	8.2	11.3	13.5	15.5	9.5	12.4	15.2	17.6	19.5	13.5	16.3	19.1	21.0	23.0	17.2	20.0	22.7	24.7	26.2	
142.2	2.0	6.9	10.3	13.3	15.5	9.7	13.4	15.8	18.7	21.0	14.5	17.8	20.7	23.3	25.6	19.5	22.0	25.4	27.9	30.0	24.3	27.3	29.8	32.3	34.0	
213.0	3.2	7.9	11.5	14.8	17.3	10.0	14.0	17.3	20.5	23.0	16.0	19.5	22.6	25.3	27.8	21.5	25.0	27.8	30.2	32.6	27.2	30.0	32.7	35.0	37.0	
285.0	3.0	7.9	11.5	14.8	17.3	10.0	14.4	17.3	20.5	23.0	16.4	20.5	23.2	25.8	27.8	22.5	26.1	28.5	31.4	33.7	28.5	31.3	34.2	36.5	38.7	
Casing temperature, 1832 deg. Fahr.																										
71.3	7.0	9.1	11.1	12.6	14.2	10.5	12.5	14.5	16.0	17.5	14.0	16.0	17.8	19.2	20.5	17.2	19.2	20.7	22.2	23.8	20.4	22.3	23.7	25.1	26.2	
142.2	10.5	13.1	15.1	17.3	18.7	15.0	17.3	19.5	21.5	22.8	19.3	21.6	23.7	25.4	26.7	23.5	25.4	27.4	29.0	30.5	27.3	29.3	31.0	32.7	33.8	
213.0	11.4	14.2	16.4	18.5	20.5	16.3	19.0	21.2	23.4	25.0	21.2	23.5	25.7	27.6	29.2	25.4	27.8	29.8	31.7	33.0	29.7	32.0	33.8	35.4	36.7	
285.0	11.0	14.2	16.5	19.0	20.8	16.3	19.5	21.6	24.0	25.6	21.5	24.2	26.4	28.5	30.2	26.3	28.8	30.9	32.9	34.5	30.8	33.7	35.1	37.0	38.3	

Comounds per sq. in. abs.

Compression pressure,
pounds per sq. in. abs.

air to 842 deg. Fahr., and a compression-pressure ratio of 15. For each assumed turbine efficiency there is a definite weight of water which must be injected to give the desired casing temperature; this weight is found by graphic methods and is given in Table III. The explosion

TABLE III.—CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITH REGENERATIVE HEATING OF THE COMPRESSED CHARGE TO 842 DEG. FAHR., WITH WATER INJECTION INTO THE COMBUSTION CHAMBER, AND WITH TURBINE-CASING TEMPERATURE 932 DEG. FAHR.

Assumptions: Compression pressure, 213.4 lb. per sq. in. abs.; back pressure at gas-turbine, 17.06 lb. per sq. in. abs.; fuel, oil; excess air, 100 per cent.

Turbine efficiency, per cent.	55	60	65	70	75	
Ratio of weight of water injected to weight of air and fuel	0.415	0.387	0.358	0.328	0.297	
Explosion pressure, lb. per sq. in. gauge	318	331	345	358	378	
Explosion temperature, deg. Fahr.	1582	1654	1737	1827	1935	
Temperature after adiabatic expansion, deg. Fahr.	518	550	585	621	657	
Initial spouting velocity, ft. per sec.	4190	4270	4360	4460	4560	
Brake thermal efficiency, per cent.						
Compressor efficiency, per cent.	55	7.7	11.3	14.8	18.5	22.3
	60	9.8	13.4	17.0	20.7	24.5
	65	11.8	15.3	19.0	22.6	26.5
	70	13.5	17.0	20.6	24.2	28.0
	75	15.0	18.6	22.3	28.0	29.7

pressures and temperatures and the spouting velocity are seen to be moderate; but the possible brake thermal efficiencies are not very promising—not more than 20 per cent. could probably be realised.

(d) **Explosion Turbine Combined with Steam-Turbine Operated from Exhaust-Heat Boiler.**—Regenerative heating of the compressed air increases the explosion and casing temperatures for a given amount of fuel burned, or, if the casing temperature is fixed, increases the amount of excess air necessary to keep down the temperatures. It consequently increases the negative work of the compressor and diminishes thereby the overall efficiency. A better method of utilizing the heat of the exhaust gases is to generate medium- or low-pressure steam with it and to use this steam in a steam-turbine. With a high exhaust temperature, a combined efficiency of boiler and economizer of 80 per cent. may be assumed as a possibility, and a steam-turbine efficiency of 20 per cent. The steam-plant efficiency is then 16 per cent., and, in most cases, the steam-turbine work will be more than is required for driving the compressor.

The gas-turbines built by Holzwarth are of this kind, with the steam-turbine driving the compressor and with all the gas-turbine work available. As the casing temperatures are high, air cooling is also employed. This case has been calculated by Stodola for casing temperatures of 1112, 1472 and 1832 deg. Fahr., and for a steam-plant efficiency of 16 per cent. Any excess of steam-turbine work over that required to drive the compressor is added to the gas-turbine work. The efficiencies attainable are given in Table IV. and are seen to be higher than the corresponding efficiencies in Table II., especially for the lower compressor efficiencies. It should again be remembered that the additional negative work imposed by the air cooling is not here taken into account, unless lower turbine efficiencies are assumed for this purpose. When air cooling is employed the exhaust-gas temperature falls, so that the attainable steam-plant efficiencies would be lowered and the brake thermal efficiencies correspondingly reduced below the values of Table IV. The plant becomes very complicated with air compressor, gas-turbine, boiler, steam-turbine, condenser, feed pump, etc.

The conditions of Table IV. demand a large excess of air. A further analysis of this cycle is given in Table V. in which the excess air is kept at 50 per cent., which would presumably ensure an explosive mixture. The casing temperature will then vary both with the amount of reheating (turbine efficiency) and with the compression ratio. This case is investigated for no compression and for compression-pressure ratios of 2 and 3. The casing temperatures are very high, ranging from 1902 to 2417 deg. Fahr., so that cooling, with its attendant losses, is necessary; no account of these losses is taken in Table V.

Three different efficiencies are tabulated. Column 3 gives the brake thermal efficiency in the case where the compression work is done by the steam-turbine and no excess work is available from that turbine. The last three columns show the steam-plant efficiencies that are necessary if the steam-turbine develops exactly the power required by the compressor; as this ranges from 5.8 to

TABLE IV.—BRAKE THERMAL EFFICIENCIES, PER CENT., OF EXPLOSION GAS TURBINES IN WHICH THE HEAT OF THE EXHAUST GASES IS USED TO GENERATE STEAM AND DRIVE A STEAM-TURBINE.

Data from Figs. 1051, 1052 and 1053 of Stodola's *Dampf- und Gas-Turbinen* (5th edition).

Assumptions: Back pressure at gas-turbine, 17·06 lb. per sq. in. abs.; fuel, producer gas with low heating value (123 B.Th.U. per cu. ft.) at 1 atm. pressure and 60 deg. Fahr.; heat loss to combustion chamber, 10 per cent. of heat of combustion; efficiency of steam boiler 80 per cent., of steam turbine 20 per cent., of steam plant 16 per cent.; compression efficiencies on isothermal basis.

Compressor efficiency, } per cent.		Turbine efficiency, per cent.																						
		55					60					65					70					75		
		55	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75			
		Casing temperature, 1112 deg. Fahr.																						
28·5		15·3	16·3	17·5	18·4	18·8	17·0	18·0	19·0	19·8	20·4	18·5	19·5	20·5	21·3	22·4	20·2	21·2	22·2	22·8	23·2			
57·0		14·7	16·4	17·9	19·1	20·3	17·2	19·0	20·5	21·5	22·9	19·7	21·5	22·8	24·1	25·1	22·3	24·0	25·3	26·5	27·5			
85·5		12·8	15·0	16·5	18·2	19·3	15·7	17·8	19·5	21·0	22·4	18·8	20·8	22·5	23·9	25·2	22·0	24·1	25·4	26·7	28·0			
114·0		11·6	13·8	16·0	17·8	19·2	15·1	17·3	19·2	21·0	22·5	18·8	21·0	22·8	24·3	25·8	22·2	24·2	26·0	27·5	28·7			
142·2		11·3	14·0	16·0	18·0	19·6	15·0	17·5	19·8	21·5	23·0	19·0	21·3	23·5	25·0	26·5	22·8	25·0	27·0	28·5	29·9			
		Casing temperature, 1472 deg. Fahr.																						
28·5		19·6	20·4	21·1	21·6	22·2	21·0	22·0	22·5	23·0	23·5	22·6	23·3	24·0	24·5	25·0	24·2	25·0	25·5	26·0	26·4			
57·0		20·1	21·1	22·5	23·4	24·1	22·3	23·5	24·5	25·5	26·3	24·6	25·8	26·7	27·7	28·5	27·0	28·0	29·0	30·0	30·7			
85·5		19·5	20·8	22·3	23·2	24·1	22·0	23·5	24·8	25·7	26·7	24·8	26·2	27·3	28·3	29·3	27·5	29·0	30·1	31·0	32·0			
114·0		19·0	20·7	22·2	23·2	24·3	22·0	23·7	25·0	26·0	27·2	25·0	26·5	27·8	28·8	29·8	28·0	29·5	30·7	31·7	32·7			
142·2		18·8	20·7	22·2	23·5	24·6	22·0	23·9	25·3	26·5	27·7	25·3	27·2	28·5	29·7	30·8	28·6	30·3	31·7	32·7	33·8			
		Casing temperature, 1832 deg. Fahr.																						
28·5		22·0	22·5	23·0	23·6	24·0	23·5	24·0	24·5	25·0	25·7	25·2	25·7	26·3	26·7	27·2	26·8	27·3	27·9	28·2	28·7			
57·0		23·2	24·0	25·0	25·7	26·5	25·0	25·7	26·5	27·3	28·0	28·5	27·8	28·8	29·5	30·2	30·8	30·1	31·0	31·7	32·5			
85·5		22·8	24·0	25·1	26·0	26·8	25·5	26·8	27·8	28·7	29·5	28·2	29·5	30·4	31·2	32·0	30·8	31·8	32·8	33·6	34·5			
114·0		23·0	24·5	25·5	26·5	27·5	25·8	27·2	28·2	29·2	29·9	28·5	29·8	31·0	32·0	32·7	31·5	32·6	33·6	34·6	35·4			
142·2		23·1	24·5	25·0	27·0	27·8	26·1	27·5	28·6	29·7	30·7	29·3	30·5	31·7	32·8	33·7	32·5	33·7	34·8	35·7	36·5			

Compression pressure,
pounds per sq. in. abs.

TABLE V.—CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITHOUT REGENERATION BUT WITH EXHAUST HEAT USED TO GENERATE STEAM AND DRIVE A STEAM-TURBINE, WITH CHARGE PRECOMPRESSION AND EXHAUST TO ATMOSPHERE.

Assumptions: Gas fuel; excess air, 50 per cent.; back pressure at turbine, 3 in. of water above exhaust pressure; temperature of charge entering combustion chamber, 59 deg. Fahr.; heat lost to walls of combustion chamber = 10 per cent. of heat of combustion.

Turbine efficiency, per cent.	Casing temperature, deg. Fahr.	Brake thermal efficiency when steam-turbine work = compressor work, per cent.	Brake thermal efficiency, per cent., including steam-turbine work (steam-plant efficiency, 15 per cent.).					Steam-plant efficiency when steam-turbine work = compressor work, per cent.				
			Compressor efficiency, per cent.					Compressor efficiency, per cent.				
			55	60	65	70	75	55	65	75		
			Ratio of compression = 3.									
50	2147	21.9	23.2	23.9	24.6	25.1	25.6	13.2	11.2	9.6		
55	2086	24.1	24.9	25.6	26.3	26.8	27.3	13.6	11.5	10.0		
60	2034	26.4	26.9	27.7	28.2	28.8	29.4	14.1	11.9	10.2		
65	1964	28.6	28.8	29.6	30.2	30.7	31.1	14.6	12.4	10.7		
	1902	30.8	30.7	31.5	32.1	32.6	33.2	15.1	12.8	11.1		
Ratio of compression = 2.												
50	2242	18.6	23.6	24.1	24.5	24.8	25.1	7.9	6.7	5.8		
55	2190	20.4	25.3	25.8	26.2	26.5	26.8	8.1	6.9	5.9		
60	2138	22.3	26.9	27.3	27.7	28.1	28.4	8.3	7.0	6.1		
65	2086	24.1	28.4	28.9	29.3	29.6	29.9	8.6	7.3	6.3		
70	2034	26.0	30.0	30.4	30.8	31.2	31.5	8.8	7.5	6.5		
No compression.												
50	2417	11.97										
55	2384	13.17										
60	2352	14.35										
65	2320	15.55										
70	2288	16.75										

15.1 per cent. it may be regarded as a possibility. Cols. 4 to 8 give the brake thermal efficiency of combined gas- and steam-turbines when the steam-plant has an efficiency of 15 per cent. It may be noted that, under these conditions of operation, precompression of the charge is not particularly valuable; for example, if compressor efficiency is 70 per cent. and turbine efficiency is 60 per cent., the brake thermal efficiency with a compression-pressure ratio of 3 is 28.8 per cent. as compared with 25.7 per cent. when no compression is used.

TABLE VA.—SUPPLEMENTARY DATA FOR TABLE V.

Ratio of compression	1	2	3
Explosion pressure, lb. per sq. in.	92	198	317
Explosion temperature, deg. Fahr.	3412	3412	3412
Temperature after expansion, deg. Fahr.	2093	1722	1526
Spouting velocity, ft. per sec.	4570	5130	5400

Supplementary data for this case are given in Table VA, which shows that the spouting velocities are high and that, in this respect, the condition with no precompression of the charge offers least difficulty. On the other hand, it is the condition with highest casing temperature and consequently of maximum cooling work losses.

(e) Explosion Turbine Combined with Steam-Turbine Operated from Exhaust-Heat Boiler with Charge Precompressed and Reduced Back Pressure.—By the addition of an exhaustor to a gas-turbine, the density of the medium in which the turbine rotates can be reduced and the windage losses consequently diminished. Spouting velocities will increase if the other conditions are unchanged, and this may necessitate a speeding up of the rotor and an increase in windage losses from this cause. The additional work of increased expansion of the explosion gases is offset by the work required to be done on the exhaustor. Since the exhaustor acts on the exhaust gases after they have been cooled (desirably to atmospheric temperature), the negative work of the exhaustor may be less than the increase in work done on the turbine, and an actual increase in brake thermal efficiency may be possible. Table VI., in which the performance of a turbine plant of this type is given, shows but little change in brake thermal efficiency

TABLE VI.—CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITH CHARGE PRECOMPRESSION AND WITH AN EXHAUSTER FOR REDUCING THE BACK PRESSURE AT THE GAS-TURBINE. THE HEAT OF THE EXHAUST GASES IS UTILIZED IN GENERATING STEAM TO DRIVE A STEAM-TURBINE.

Assumptions: Oil fuel; excess air, 50 per cent.; temperature of charge entering combustion chamber, 59 deg. Fahr.; back pressure at gas-turbine, 2.84 lb. per sq. in. greater than exhaustor pressure; ratio of compression = ratio of exhaustion; temperature of water fed to boiler, 59 deg. Fahr.; heat lost to walls of combustion chamber = 10 per cent. of heat of combustion.

Turbine efficiency, per cent.	Casing temperature, deg. Fahr.	Brake thermal efficiency when steam-turbine work = compressor work, per cent.	Brake thermal efficiency, per cent., including steam-turbine work (steam-plant efficiency, 15 per cent.).					Steam-plant efficiency when steam-turbine work = compressor + exhaustor work, per cent.				
			Compressor efficiency, per cent.					Compressor efficiency, per cent.				
			55	60	65	70	75	55	65	75	55	65
Ratio of compression = ratio of exhaustion = 1.5.												
50	2234	17.4	22.2	22.7	23.1	23.5	23.8	8.4	7.1	6.2		
55	2190	19.1	23.7	24.2	24.7	25.0	25.3	8.6	7.3	6.3		
60	2145	20.9	25.2	25.7	26.2	26.5	26.8	8.8	7.5	6.5		
65	2100	22.6	26.6	27.1	27.5	27.9	28.2	9.0	7.6	6.6		
70	2052	24.3	28.1	28.6	29.0	29.4	29.7	9.3	7.9	6.8		
Ratio of compression = ratio of exhaustion = 2.												
50	2124	21.6	21.5	22.4	23.1	23.7	24.3	15.0	12.9	11.2		
55	2064	23.7	23.3	24.2	24.9	25.5	26.1	15.0	13.4	11.5		
60	2006	25.9	25.1	26.0	26.7	27.4	27.9	16.3	13.8	11.9		
65	1944	28.1	27.0	27.9	28.6	29.3	29.8	16.8	14.2	12.4		
70	1890	30.3	28.9	29.7	30.5	31.1	31.5	17.4	14.8	12.8		
Ratio of compression = ratio of exhaustion = 3.												
50	1988	26.6	19.6	21.0	22.1	23.1	24.0	26.0	22.0	19.1		
55	1912	29.3	21.9	23.2	24.4	25.4	26.3	27.2	23.0	20.0		
60	1832	31.9	24.1	25.5	26.7	27.7	28.5	28.4	24.1	20.8		
65	1764	34.6	26.4	27.8	28.9	29.8	30.8	29.8	25.2	21.8		
70	1692	37.3	28.6	30.0	31.2	32.2	33.0	31.3	26.4	22.9		
Ratio of compression = ratio of exhaustion = 1. (No compression or exhaustion).												
50	2421	10.4			22.4							
55	2396	11.5			23.3							
60	2369	12.5			24.1							
65	2342	13.5			25.0							
70	2315	14.5			25.8							

as compared with Table V., which is the generally similar case without exhauster ; any advantage is to be looked for principally in reduction of windage loss. The complexity of the plant is even greater than in the preceding case in consequence of the addition of the exhauster. With fuels containing sulphur there will be diluted sulphuric acid in the cooled exhaust gases, and corrosion of the economizer and the exhauster would have to be guarded against. From the last three columns of Table VI. it is apparent that the steam-turbine would not be able to drive both the compressor and the exhauster for ratios of compression (= ratio of exhaustion) greater than 2.

Table VIA contains supplementary data for the conditions of Table VI.

TABLE VIA.—SUPPLEMENTARY DATA FOR TABLE VI.

Ratio of compression = ratio of exhaustion	.	1	1.5	2	3
Explosion pressure, lb. per sq. in.	.	90	143	194	299
Explosion temperature, deg. Fahr.	.	3352	3352	3352	3352
Temperature after expansion, deg. Fahr.	.	2150	1760	1520	1229
Spouting velocity, ft. per sec.	.	4480	5110	5450	5810

Further calculations for this cycle are given in Table VII., with the following differences in conditions from Table VI. Gas fuel instead of oil fuel ; back pressure at turbine, 3 inches of water above exhauster pressure instead of 2.84 pounds per square inch ; calculation made only for ratio of compression pressure = ratio of exhauster = 1.5. The resulting brake thermal efficiencies are improved by the reduction in back pressure ; the windage losses would also be reduced.

(f) **Constant-Pressure-Combustion Turbine with Regeneration.**—If no cooling device is used the maximum permissible casing temperature may be taken as 932 deg. Fahr. and the corresponding regenerative heating to 842 deg. Fahr. These are the same as the conditions assumed for the explosion cycle of Table I. The calculated performance of this cycle is given in Table VIII., for the same compression ratios as in Table I.; the efficiencies obtained are not very different in the two cases and are higher for the explosion cycle except at low compression-pressure ratios. The casing temperature

TABLE VII.—CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITHOUT REGENERATION BUT WITH EXHAUST HEAT USED TO GENERATE STEAM AND DRIVE A STEAM-TURBINE, AND WITH EXHAUSTER TO REDUCE BACK PRESSURE ON GAS-TURBINE.

Assumptions: Ratio of compression = ratio of exhaustion = 1.5; temperature of charge entering combustion chamber, 59 deg. Fahr.; fuel, gas; back pressure at turbine, 3 in. of water above exhauster pressure; excess air, 50 per cent.; heat loss to walls of combustion chamber = 10 per cent. of heat of combustion; temperature of water fed to boiler, 59 deg. Fahr.

Turbine efficiency, per cent.	Casing temperature, deg. Fahr.	Brake thermal efficiency when steam-turbine work = compressor work, + exhauster work, per cent.	Brake thermal efficiency, per cent., including excess steam-turbine work (steam-plant efficiency, 15 per cent.),					Steam-plant efficiency when steam-turbine work = compressor + exhauster work, per cent.				
			Compressor efficiency, per cent.					Compressor efficiency, per cent.				
			55	60	65	70	75	55	65	75		
50	2213	19.3	23.7	24.2	24.6	25.0	25.3	9.02	7.64	6.62		
55	2164	21.3	25.3	25.8	26.3	26.6	27.0	9.28	7.86	6.80		
60	2110	23.3	26.9	27.4	27.9	28.3	28.6	9.56	8.09	7.00		
65	2050	25.2	28.6	29.1	29.5	29.9	30.2	9.84	8.34	7.22		
70	2002	27.2	30.2	30.7	31.2	31.6	31.9	10.15	8.60	7.44		

TABLE VIII.—CALCULATED PERFORMANCE OF CONSTANT-PRESSURE-COMBUSTION GAS-TURBINES WITH REGENERATIVE HEATING OF THE COMPRESSED CHARGE TO 842 DEG. FAHR., AND TURBINE-CASING TEMPERATURE 932 DEG. FAHR.

Assumptions: Back pressure at gas-turbine, 17.00 lb. per sq. in.; gas fuel; heat loss to walls of combustion chamber, 10 per cent. of heat of combustion; compression efficiencies on isothermal basis.

Compression pressure, lb. per sq. in.	71.12					142.24					213.36					284.48				
	60	65	70	75	80	60	65	70	75	80	60	65	70	75	80	60	65	70	75	80
Turbine efficiency, per cent.	1240	1272	1308	1344	1380	1382	1438	1494	1551	1612	1472	1531	1590	1660	1742	1531	1603	1683	1765	1850
Maximum combustion temperature, deg. Fahr.	713	736	761	786	811	608	642	676	712	750	550	585	622	664	709	505	547	590	635	682
Temperature after adiabatic expansion, deg. Fahr.	1580	1442	1335	1222	1127	1113	994	890	799	717	940	830	732	640	572	830	724	634	556	495
Excess air, per cent.	2062	2702	2739	2770	2800	3237	3290	3340	3387	3438	3535	3595	3653	3715	3780	3720	3790	3870	3950	4032
Spouting velocity, ft. per sec.	Brake thermal efficiency, per cent.																			
	55	3.1	9.2	6.1	12.5	0.4	8.0	15.3	1.4	9.02	18.0	8.0	15.0	23.5	5.5	12.8	19.3	25.5	5.5	13.6
	60	2.8	9.0	14.7	5.5	12.2	18.1	7.1	14.2	20.8	8.0	15.0	23.5	5.5	13.6	20.9	28.1	2.4	10.9	18.4
	65	1.7	8.2	14.1	19.4	4.0	11.1	17.4	22.9	2.6	10.7	17.5	23.7	29.5	7.5	15.5	22.6	29.2	33.6	
	70	0.2	6.8	12.9	18.5	23.5	1.5	9.3	16.0	21.8	26.9	7.7	15.4	21.7	27.4	33.0	7.5	15.5	22.6	29.2
	75	4.8	11.1	16.9	22.2	27.0	6.4	13.7	20.1	25.6	30.4	7.7	15.4	21.7	27.4	33.0	7.5	15.5	22.6	29.2

limits the maximum combustion temperature and thereby limits the amount of fuel that may be burned. The excess air consequently is very high, ranging from 485 to 1580 per cent. This results in a high ratio of the negative work of compression to the positive work of the turbine and thereby results in a low brake thermal efficiency.

(g) **Constant-Pressure-Combustion Turbine with Regeneration and with Cooling of the Buckets by Steam Jets.**—The constant-pressure-combustion turbine without cooling (Table VIII.) is necessarily of low efficiency. Air cooling entails considerable losses. Another possibility in cooling is to generate steam by the exhaust gases and to expand this steam through nozzles on to the gas-turbine, which thus serves both as a gas- and a steam-turbine. The steam will be used inefficiently in this case as it will be operating as a non-condensing turbine, but, as it discharges as wet steam at 212 deg. Fahr., it will serve as an excellent cooling medium and will reduce the casing temperature.

Two cases of this cycle have been analysed. In both of them oil fuel is used with 100 per cent. excess air, a bucket peripheral velocity of 800 feet per second is assumed, the back pressure at the turbine is taken as 17.06 pounds per square inch abs., and the exhaust gases are supposed to be cooled to 180 deg. Fahr. above the steam temperature in the exhaust-heat boiler and then to go through a regenerator. It is also assumed in both cases that the 10 per cent. of the heat of combustion which goes to the combustion-chamber walls is used in evaporating water in the jackets around it. The compression pressure is 101 pounds per square inch abs. The steam-nozzle velocity coefficient is taken as 0.95. The combustion temperature is 2370 deg. Fahr.

(i) In the first case it is assumed that steam is generated at such pressure that it has the same spouting velocity as the gases (3600 feet per second) so that it can be efficiently utilized by the turbine. The gas- and steam-turbines are assumed to have 70 per cent. efficiency, the compressor 60 per cent.

Computations on this basis yield the following results :

Weight of steam generated per lb. of exhaust gas, lb.....	0·214
Pressure of steam, lb. per sq. in. abs.....	1070
Gas temperature after adiabatic expansion, deg. Fahr.	1441
Casing temperature of gas alone, deg. Fahr.	1745
Quality of steam after adiabatic expansion.....	0·751
Quality of steam after reheating in turbine.....	0·845
Casing temperature with combined gas and steam, deg. Fahr...	1269
Brake thermal efficiency, per cent.....	20·5

(ii) The steam pressure necessary to give the required spouting velocity in case (i) is very high (1070 pounds per square inch abs.). It seemed desirable to make computations also for the case in which the steam acts only on the second row of buckets and consequently has a spouting velocity of 1800 feet per second. It was assumed for this case that the steam-turbine efficiency was 70 per cent., gas-turbine efficiency 75 per cent., compressor efficiency 65 per cent. The computed results follow.

Weight of steam generated per lb. of exhaust gas, lb.....	0·314
Pressure of steam, lb. per sq. in. abs.....	45·5
Gas temperature after adiabatic expansion, deg. Fahr.	1441
Casing temperature of gas alone, deg. Fahr.....	1697
Quality of steam after adiabatic expansion.....	0·946
Quality of steam after reheating in turbine.....	0·97
Casing temperature with combined gas and steam, deg. Fahr...	1184
Brake thermal efficiency, per cent.....	19·9

It will be seen that the cases calculated above yield casing temperatures of 1269 and 1184 deg. Fahr. respectively, which are too high to be practicable. With larger excess air this condition could be remedied, but at the cost of reduced brake thermal efficiencies. These efficiencies are already quite low.

(h) **Constant-Pressure-Combustion Turbine with Regeneration and with Steam Injection into the Combustion Space.**—Another method of using the steam generated in an exhaust-gas boiler is to discharge it directly into the combustion space. It is assumed to be generated at 10 per cent. above the compression pressure. As no additional cooling is used, the casing temperature is limited to 932 deg. Fahr. The exhaust gases are assumed cooled to 572 deg. Fahr. in the boiler, the compressed air is heated by the regenerator to 392 deg. Fahr., and the feedwater is heated by the exhaust gases to 122 deg. Fahr. In addition to the heat from the exhaust gases,

the 10 per cent. of the heat of combustion which goes to the jacketed walls of the combustion chamber is assumed to be used in generating steam. The back pressure at the turbine is 17.06 pounds per square inch abs.

TABLE IX.—CALCULATED PERFORMANCE OF CONSTANT-PRESSURE-COMBUSTION TURBINE WITH STEAM INJECTION INTO THE COMBUSTION SPACE, WITH REGENERATION TO 392 DEG. FAHR. AND WITH TURBINE-CASING TEMPERATURE OF 932 DEG. FAHR.

Assumptions : Oil fuel; excess air, 200 per cent.; steam pressure, 10 per cent. greater than compression pressure; back pressure, 17.06 lb. per sq. in. abs.; steam generated by cooling exhaust gases to 572 deg. Fahr. and by 10 per cent. of heat of combustion going through combustion-chamber walls; feedwater heated to 122 deg. Fahr. by exhaust gases.

Compression-pressure ratio	40	17.6	10
Combustion temperature after steam admixture, deg. Fahr.	1571	1557	1550
Temperature after adiabatic expansion, deg. Fahr.	369	565	730
Turbine efficiency ratio, per cent.	54.9	64.2	76.1
Spouting velocity, ft. per sec.	4210	3840	3500
	Brake thermal efficiency, per cent.		
	55	5.8	13.8
	60	9.5	16.8
Compressor efficiency, per cent.	65	3.3	12.6
	70	6.7	15.2
	75	9.7	17.5
			23.2

This case was calculated for an assumed excess of air. This gives a definite combustion temperature and also, for each compression-pressure ratio, a definite temperature at the end of adiabatic expansion. With a casing temperature limited to 932 deg. Fahr. this fixes the amount of reheating in the turbine and therefore the turbine efficiency. In Table IX. the turbine efficiencies are those which must be reached if the casing temperature is not to exceed the stated value.

When calculated for 100 per cent. excess air, the turbine efficiencies must be over 85 per cent. if the casing temperature is to be kept down to 932 deg. Fahr., even for very high compression pressures.

With 200 per cent. excess air, the ratio of weight of steam to weight of products of combustion is 0.144 and the temperature of combustion before the injection of the

steam is 1832 deg. Fahr. The calculated efficiencies and other data are given in Table IX.

An examination of this table shows that with turbine efficiencies of 54·9, 64·2, and 76·1 per cent. the necessary compression-pressure ratios, if the casing temperature is to be kept down to 932 of abs., are 40, 17·6, and 10, respectively. With turbine efficiency of 76·1 per cent. and compressor efficiency of 75 per cent. (both values beyond existing possibilities) the brake thermal efficiency is only 23·2 per cent.

Conclusions.—The advantages which the steam-turbine has over the reciprocating engine are chiefly high rotative speed, absence of cylinder oil, large power per unit, low weight, low attendance cost, compactness, simplicity, small weight, and higher efficiency. The gas-turbine also has the first three items as probable advantages, but the other items cannot be definitely claimed, at any rate in so great a degree as in the steam-turbine. A simple gas-turbine without use of the exhaust heat would have too low an efficiency to be practical. If a regenerator is added, but no provision for cooling is made, the operating conditions have to be such that the efficiencies obtainable are very low. If cooling is provided the weight and complication of the plant increase. With the explosion turbine (which appears to be the most practical in view of the lower compression pressures required and the possibilities of cooling) the use of valves in the combustion chamber introduces a type of complication from which the steam-turbine is free and which may be expected to be the source of much trouble.

We are therefore forced to the conclusion that, even with the increase in compressor and turbine efficiencies which may be expected to result from further developments, it seems highly improbable that brake thermal efficiencies as high as 25 per cent. could be obtained. The maximum that has been claimed up to the present is only 13 per cent., a figure which compares ill with brake thermal efficiencies of about 34 per cent. obtained with Diesel engines and about 38 per cent. with the Still engine.

THE EXHAUST-GAS-TURBINE¹

WHILE little success has attended efforts to build a gas-turbine capable of dealing with the total available temperature and pressure range of gases heated by internal combustion, there would appear to be a field for the employment of this type of prime-mover in the utilization of the waste heat in the exhaust gases of an ordinary reciprocating engine. The most notable achievement in this direction is the application by Professor Rateau of the exhaust-gas-turbine to the aircraft motor. It may be of interest therefore to inquire into the peculiar circumstances which led to its evolution and subsequent commercial success.

The Need for the Exhaust-Gas Airplane Turbine.

The exigencies of war conditions called for speedier and speedier planes rising to ever greater heights, and, as will be shown, there is considerable advantage to be derived under normal conditions of commercial service from high altitude flight. Hence the problem has really two aspects : the derivation of greater power from the engine with a minimum increase in weight and complexity, and the amelioration of the low temperature and density conditions of the upper atmosphere.

The Advantages of 'High Ceiling.'—The essential difference between the circumstances of air locomotion, and those of other forms with which we are familiar, is that, whereas the latter are determined by factors such as the resistance of water, or air of nearly uniform density, which vary according to some power of the speed, the aeroplane is at liberty to select its own medium if it can rise sufficiently high.

¹ The author is indebted for information concerning the exhaust-gas-turbine to the original paper of Professor Rateau, published in the *Proc. Inst. of Mech. Eng.* (London).

The time taken by a transatlantic liner to cross from Europe is a compromise between the demand on the one hand for a rapid passage, and the cost, based upon fuel consumption, on the other. In ocean travel, for example, there comes a point comparatively quickly where the gain in time is out of all proportion to the expenditure involved, the resistance to the motion of the hull obeying the basic fluid law :

$$R \propto \rho v^n \quad . \quad . \quad . \quad (1)$$

n having the value two (approximately) and ρ being a density factor.

This, of course, is true of the aeroplane also ; but ρ diminishes with an increase in elevation and may thus be made as small as possible. The proposition may be stated generally in the form :

$$R = k\rho v^2 \quad . \quad . \quad . \quad (2)$$

where k comprises a function of the incidence of the wings plus a constant for the fuselage, etc.

The work done by the airscrew at a given velocity v is

$$W_{AS} = Rv \text{ ft. lb. per sec.} \quad . \quad . \quad (3)$$

Hence we have from equation (2) above :

$$Rv = k\rho v^3 \quad . \quad . \quad . \quad (4)$$

and thus we see that

$$v \propto \frac{1}{\sqrt[3]{\rho}} \quad . \quad . \quad . \quad (5)$$

the work done remaining constant.

Suppose, for example, to take a specific case, that an aeroplane can fly at 170 miles per hour near the ground. What speed would it have at such an elevation that ρ had only one-fourth of its value at the earth's surface ?

We have from (5) above that

$$\frac{v_o}{v_h} = \frac{\rho_h^{\frac{1}{3}}}{\rho_o^{\frac{1}{3}}} ; \quad v_h = v_o \left(\frac{\rho_o}{\rho_h} \right)^{\frac{1}{3}} .$$

$$\therefore \quad v_h = 170 (4)^{\frac{1}{3}} = 270 \text{ miles per hour.}$$

We cannot, of course, practically neglect the increase in weight due to the larger wing area and other complications which may be necessary to reach the desired height ; but sufficient has been said perhaps to indicate the advantages of what is termed ' high ceiling.'

Pressure and Temperature in the Upper Atmosphere.—We turn, therefore, to a consideration of atmospheric density, in its relation to elevation, in order to find whether the change in the magnitude of ρ suggested above may be expected at an attainable elevation.

The curve relating pressure and elevation in the atmosphere has a negative slope. In order to fix the ideas, consider a vertical column one square foot in cross-section. At any particular height

$$\rho = - \frac{dp}{dh}$$

where dp represents the change in pressure in pounds per square foot, and dh represents the corresponding increment of the elevation, reckoned from the ground level.

This becomes

$$dp = - dh\rho \quad . \quad . \quad (6)$$

Now, if the temperature of the air were constant, ρ would be proportional to p and we should have

$$\rho = \rho_o \frac{p}{p_o}$$

and hence

$$dp = - dh\rho_o \frac{p}{p_o} \quad . \quad . \quad (7)$$

and therefore also

$$dh = - \frac{dp}{\rho_o} \frac{p_o}{p}$$

where ρ_o = density at ground level

p_o = pressure at ground level

p = pressure at any given elevation.

Integrating between assigned limits,

$$\int_0^H dh = -\frac{p_o}{\rho_o} \int_{p_o}^{p_h} \frac{dp}{p}$$

which gives

$$H = -\frac{p_o}{\rho_o} \log_e \frac{p_h}{p_o} \quad . \quad . \quad (8)$$

and enables us to determine the pressure p_h at a given height H . The numerical value of the factor $\frac{p_o}{\rho_o}$ is easily found as follows :

$$\begin{aligned} p_o &= 14.7 \times 144 \text{ pounds per square foot} \\ \rho_o &= 0.08071 \text{ pounds per cubic feet at } 32 \text{ deg. Fahr.} \end{aligned}$$

$$\frac{p_o}{\rho_o} = \frac{14.7 \times 144}{0.08071} = 26230.$$

The expression may now be written

$$\begin{aligned} H &= -26230 \times 2.303 \log_{10} \frac{p_h}{p_o} \quad . \quad (9) \\ &= -60400 \log_{10} r \end{aligned}$$

where r is the ratio in which the pressure falls, between a point on the surface and another point H feet vertically above it.

This is generally known as the Halley formula, and is not applicable in this form to the lower strata of the atmosphere due to the change in temperature.

Balloon and other observations show that above some six or seven miles the temperature remains sensibly constant at about -65 deg. Fahr., after a more or less steady fall of about 1 deg. Fahr. per 333 feet change in elevation.

When modified empirically equation (9) becomes

$$H = -60400 \left\{ 1 - \frac{H}{285000} \right\} \log_{10} r \quad . \quad (10)$$

The factor $\left\{ 1 - \frac{H}{285000} \right\}$, it will be observed, produces a linear change in the constant 60400 with

increase in H , and equation (10) should obviously not be used beyond the boundary between the stratosphere and lower atmosphere.

Selecting, then, a point at an elevation of 34,320 feet, let us calculate by means of equation (10) the pressure at this height. Then, knowing the temperature, we are in a position to determine the density ρ .

$$34320 = -60400 \left\{ 1 - \frac{34320}{285000} \right\} \{ \log_{10} p - \log_{10} p_0 \}$$

$$34320 = 60400 (1 - 0.122) \{ \log_{10} p_0 - \log_{10} p \}$$

$$\frac{34320}{60400 \times 0.878} - \log_{10} 14.7 \times 144 = -\log_{10} P$$

$$\log_{10} P = -0.645 + 3.3255$$

$$\begin{aligned} P &= 479.2 \text{ pounds per square foot} \\ &= 3.33 \text{ pounds per square inch.} \end{aligned}$$

Now, the temperature of the atmosphere at this elevation is about -43 deg. Fahr., and hence the density becomes

$$\begin{aligned} \rho_{34320} &= \frac{3.33}{14.7} \times \frac{460}{417} \rho_0 \\ &= 0.250 \rho_0 \end{aligned}$$

or one-fourth that at ground level.

Thirty-four thousand three hundred feet is five thousand feet higher than Mount Everest and beyond the limit where respiration is supported in safety. The use of the stratum at this height would doubtless entail the provision of an airtight cabin for passengers and crew.

The Influence of Low Atmospheric Density upon the Motor.—We come now to the consideration of the influence of the change in atmospheric density upon the motor. If the carburetter produces a constant ratio of petrol and air, the I.H.P. would then be proportional to the weight of air entering the cylinders per minute. Other factors remaining constant, the power developed will diminish with ρ and the available power at the screw will diminish more rapidly, since there is a nearly constant

demand by valve gear, oil and water pumps, frictional resistances, etc. for power.

The torque applied at the screw is given approximately by the expression $T = K \left(\frac{\rho}{\rho_o} - \gamma \right)$, where K is a constant for the individual motor and γ equals a constant having a numerical value of about 0.12.

There is thus a very definite limit set to the height to which the plane can climb, since no increase in the engine speed could ultimately prevent the torque becoming zero when $\frac{\rho}{\rho_o} = 0.12$.

It will also be apparent that, if the density of the inhaled air can be maintained constant, the limit of height will be fixed by the mechanical strength of the air-screw, for, of course, as the atmospheric density diminishes, the angular velocity of the blades must be increased.

Early Designs.—During the war several of the belligerent nations, including France, England, and Germany, experimented with devices designed to maintain the density of the induction air. These early designs embodied a compressing fan, driven through a change gear directly by the engine. These proposals appear to have suffered from their complexity and the inherent disadvantage of reducing the net available power by an appreciable amount.

The turbo-compressor designed by Professor Rateau (France) utilized the waste heat in the exhaust gases in a direct-coupled single-velocity-stage turbine and centrifugal fan. This arrangement avoided at once both the objections above referred to, and while still in its experimental stages met with no small measure of success.

The Power Available in the Waste Gases.—It may be of interest to consider what amount of energy may be expected from this source and to show that, when unavoidable losses have been met, the balance will suffice to produce the desired effect. Professor Rateau states that the waste gases from an aviation motor, when running under ordinary conditions at ground level,

have an observed temperature after partial cooling of 1800 deg. Fahr. abs.

If the temperature of the atmosphere in which the motor is at work be of the order of 450 deg. Fahr. abs. the volume of the waste gases will be more than four times the volume of the air entering (there being an addition of some 5 or 6 per cent. of petrol); and consequently the former will be able to perform more than four times the amount of work necessary for compression, when expansion on the one hand and compression on the other take place between the same pressure limits.

For, let

P_1 = pressure reached by compression
= pressure of exhaust gases.

P_2 = pressure of the atmosphere
= pressure after expansion of the waste gases.

V_1 = volume of compressed air reckoned on any convenient basis.

V_2 = volume of induction air before compression by the fan.

Then $V'_2 = 4V_2$ = volume of waste gas after expansion,
and $V'_1 = 4V_1$ = volume of waste gas before expansion, assuming a law of the form

PV^n = a constant to hold and n to have the same value in both cases.

For compression, we have

$$\int_{P_2}^{P_1} V dP = \frac{n}{n-1} (P_1 V_1 - P_2 V_2)$$

and for expansion,

$$\begin{aligned} \int_{P_2}^{P_1} V dP &= \frac{n}{n-1} (P_1 \times 4V_1 - P_2 \times 4V_2) \\ &= \frac{4n}{n-1} (P_1 V_1 - P_2 V_2) \end{aligned}$$

If we can produce a combination of fan and turbine (and this is not difficult) having an overall efficiency of 25 per cent. it will be apparent that the required power

may be obtained without introducing an appreciable back pressure upon the piston.

Actual Design.—The actual design provided for the regulation of the flow of the exhaust gas by means of a butterfly valve and made possible the precise regulation of the degree of compression by causing a greater or less volume to pass through the expansion nozzles. The rotor, as far as could be gathered, resembled the De Laval type and was mounted, together with a simple radial-bladed fan, on the same flexible spindle.

The selection of suitable material for this spindle presented peculiar difficulties, due to the wide range of temperature and high rotative speed (about 30,000 revolutions per minute) to which it was subjected in service.

An intercooler placed in the fan-motor circuit reduced the temperature of the induction air which was warmed by compression in the fan.

REFRIGERATION

THE cycle of the refrigerating machine for a gas and for a liquid and its vapour has already been discussed. It remains therefore to present one or two characteristic industrial applications. The air machine is relatively bulky and inefficient and is rarely used. Of the other two media in most common use, ammonia is largely preferred for land service, while carbon dioxide is often used at sea. The former has several advantages over the latter from the point of view of efficiency and convenience, the pressures for the desired temperature range being moderate, but there is greater objection to the escape of ammonia into a confined space. Sulphur dioxide finds much favour for small domestic plants.

The Process of Refrigeration.—The simplest form of refrigerating apparatus would consist of two parts, a tank of the cooling medium and an evaporator or congealer, as shown in fig. 60. With this apparatus the refrigerating medium is allowed to escape from the tank into the evaporator or congealer as fast as the coils therein are capable of giving up sufficient heat to vaporize the liquid into a gas. This latter process takes place, of course, at the temperature corresponding to the pressure within the coils. For example, if the pressure within the coils were that of the atmosphere and the medium were ammonia, evaporation would take place at a temperature of about -27 deg. Fahr. If, now, the resulting gas or vapour were allowed to escape into the atmosphere there would be a total loss of refrigerant and the supply would have to be maintained by adding additional tanks. This, however, would be impracticable, on

account of the cost of the medium. In order to recover the gas and reconvert it into liquid, over and over again, additional apparatus is required.

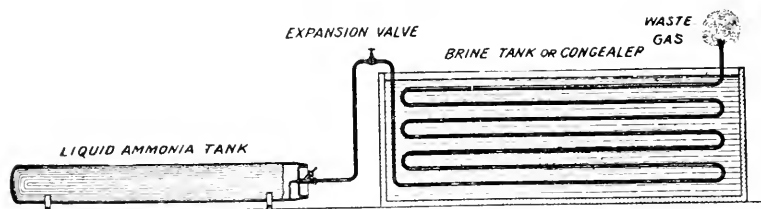


FIG. 60.—Simple Refrigerating Apparatus.

Simple Refrigerating Plant.—The simplest form of refrigerating apparatus embodying this additional equipment would consist of three principal parts shown in fig. 61.

A is an evaporator, or, as it is sometimes called, a

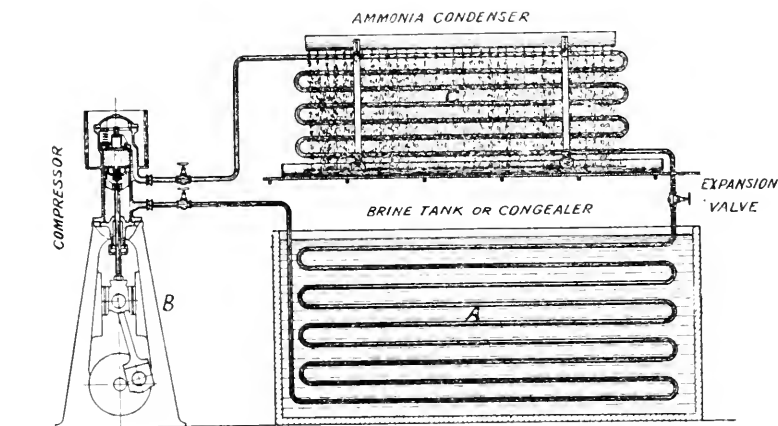


FIG. 61.—Essential Elements of Refrigerating Plant.

congealer, in which the liquid refrigerating medium is vaporized due to the absorption of heat from the surrounding brine.

B is a combined suction and pressure pump called the compressor, which transfers the gas from the evaporator coils as fast as it is formed.

C is a liquefier, or, as it is commonly called, a con-

denser, into which the gas is discharged at comparatively high pressure.

Under the action of the cooling water the heat of vaporization is abstracted and the gas condenses. For example, taking the case of ammonia once more, if the pressure within the condenser were 169.5 pounds per square inch abs., the temperature at which condensation would take place would be 86 deg. Fahr.

Referring again to fig. 61, the apparatus is first charged with the necessary quantity of cooling medium which is stored in the lower part of the condenser *C*. A small cock or expansion valve in the pipe leading to the congealer or brine tank *A* is opened slightly, thus allowing the liquid to pass into the evaporator coils. These coils perform the same office as a tube or flue in a steam boiler, and may, with equal propriety, be named the heating surface. The amount of water converted into steam in a boiler depends on the number of square feet of heating surface, the temperature of the fire, and the resulting pressure which the steam exerts—in other words, upon the total heat of the steam. The same is true of the capacity of the heating or heat-robbing surface of the coils in the evaporator.

The heat is transmitted through these coils, being taken from the substance surrounding them, which may be brine, or merely the air of the room in which the coils are placed. This heat causes the refrigerating medium to boil and creates a vapour just as water on boiling gives off steam. The surrounding substance thus parts with an amount of heat equivalent to that transferred to the cooling medium, where it is taken up and absorbed in proportion to the number of pounds of liquid evaporated. The quantity of liquid evaporated is under the control of the expansion valve leading from the condenser.

As the gas begins to form in the evaporator the compressor pump *B* is set in motion at such speed as to carry away the gas as fast as it is formed. It is then discharged into the condenser under such pressure and temperature as will bring about condensation and restore the gas to its liquid state, ready again to pass through the expansion valve. This constitutes the complete cycle, which is

continuous so long as the compressor is kept in operation and the proper quantity of water is circulated over the condenser. The condenser absorbs the heat equivalent of the work done in the compressor and the heat that the refrigerant has absorbed in the evaporator.

Whatever be the nature of the medium the principle of the action in all plants of this type is the same. The properties of the particular agent employed, of course, and the manner of its use, affect the efficiency and, consequently, the economy of operation. As ammonia is the medium most commonly employed at the present time, consideration will be given to a characteristic plant employing the ammonia compression system.

Modern Ammonia Refrigerating Plant.—Fig. 62 shows an ammonia plant manufactured by Messrs. Frick Bros., Waynesboro', Pennsylvania, U.S.A.

Liquid ammonia is charged into the system by means of the valve *M* and charging connection valve *N*. After all leaks have been stopped and air extracted from the system by the compressor functioning as a pump, water is started over the ammonia condenser and through the cooling jackets of the machine. The latter is then started and the valve on the ammonia supply drum opened slightly to give about 15 pounds per square inch gauge suction pressure on the low side. When one drum is empty, another is substituted until the system is charged, that is, until the gauge glass on the liquid receiver shows the latter to be about three-fourths full. When all is in order, the action proceeds under the control of the expansion valve shown inside the refrigerating chamber. At this point a drop in pressure from the upper to the lower pressure limit takes place, and the liquid, partly vaporized by the throttling action of the expansion valve, boils to a vapour at the lower temperature limit inside the ammonia expansion coil located in the refrigerator.

The gas then passes by way of the suction line through the suction stop valve *B* and by-pass valve *C'*, to the strainer *I* and thence to the compressor. After compression the gas passes through the discharge stop valve

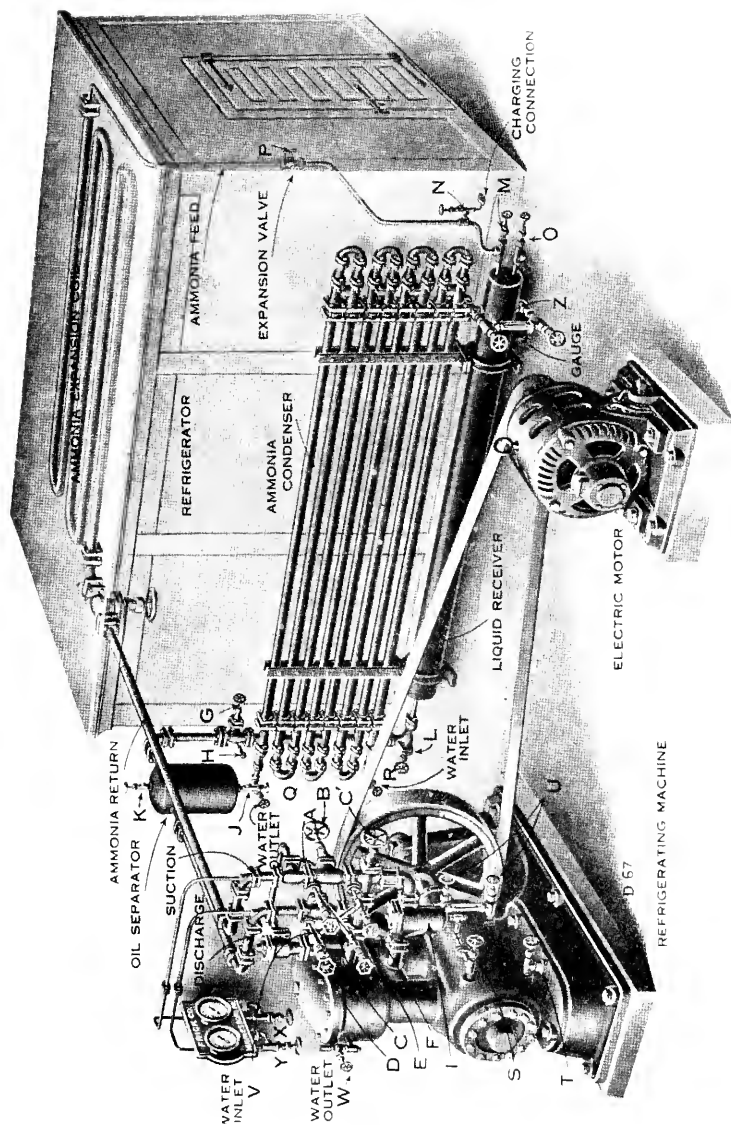


FIG. 62. Frick Ammonia Refrigeration Plant.

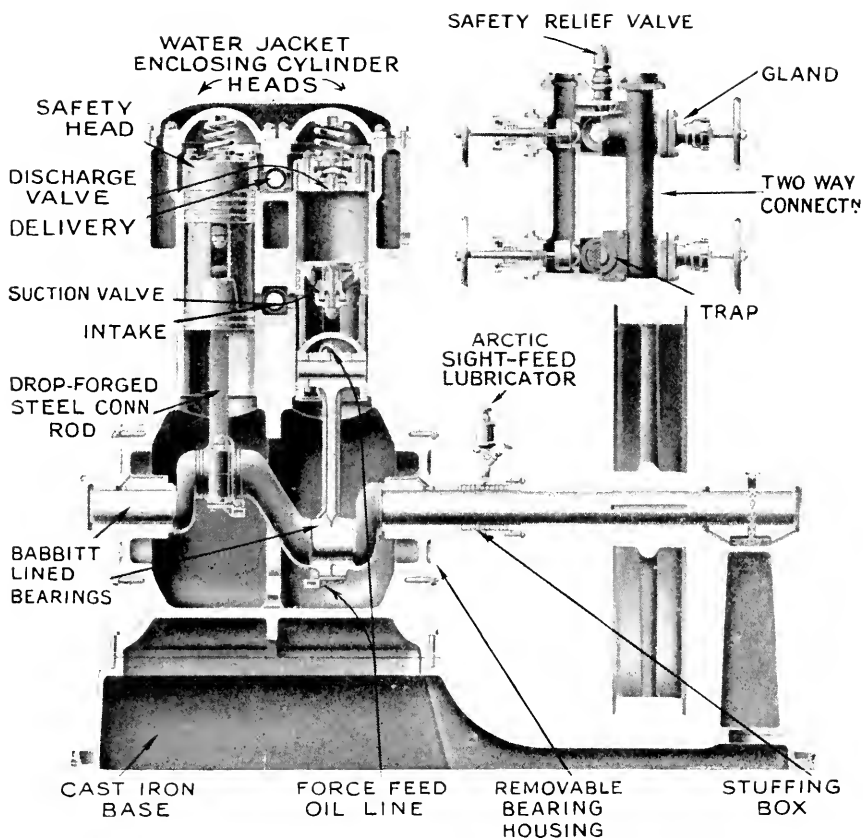


FIG. 63. Frick Ammonia Compressor.

A and by-pass valve *C* to the oil separator and thence to the condenser through the stop valve *G*. *K* and *J* are purge and drain valves respectively, mounted on the oil separator. *S* and *T* are oil connections for the crank chamber. *D*, *E*, and *F*, by-pass valves, better seen in fig. 63. *U* indicates the oil gauge on the crank chamber, and *L* controls the admission of ammonia to the receiver.

‘Frick’ Ammonia Compressor.—Referring to fig. 63, ammonia gas which may be wet, dry, or superheated, depending upon the conditions of working, enters by way of the intake, being sucked in under the action of the piston on the up stroke.

The gas finds its way around the annular space between the body of the piston and the walls of the cylinder and into the hollow trunk piston. On the down stroke the suction valve set in the piston head lifts owing to the vacuum produced by the piston in its descent. The gas is thus transferred to the upper side of the valve and fills the cylinder. In order to minimize the throttling effect of the suction valve the latter is balanced by the pressure of a spring and its own weight, and tends to ‘float’ in the guides. On the up stroke the ammonia is compressed to the pressure of the delivery line and is then forced through the discharge valves on its way to the condenser. The ‘safety head’ shown in the illustration should not be confused with the discharge valves which are placed within it. Since the clearance is practically nil, the coil spring located on top of the head acts as a safety device, permitting the head to lift in the event of liquid ammonia entering the cylinder in the form of a ‘slug.’ The function of the water jacket, through which a constant stream of water circulates, is, of course, to remove heat generated by compression, thus keeping the machine cool and assisting the condenser in the work of heat extraction. In the figure showing pipe connections, if the line on the left is connected to the high-pressure side of the system the upper valve would normally be open and the lower closed. For the right-hand line connected to the low-pressure side the reverse is true. By reversing the direction of flow the

high-pressure side may be pumped out in the event of a leak, and the ammonia in the system thus conserved in the low-pressure side until the necessary repairs can be effected.

Application to the Manufacture of Ice.—In order to produce ice from water, the water must first be cooled to 32 deg. Fahr. and then an amount of heat equal to the latent heat of ice—namely, 144 B.Th.U.—extracted. The unit of refrigeration in the United States is that amount of heat which must be abstracted from 2000 pounds of water at 32 deg. Fahr. to produce 2000 pounds of ice at the same temperature. The unit of refrigeration is accordingly 2000 times 144 equals 288,000 B.Th.U., this being called 1-ton refrigeration. The specified time in which this amount of refrigeration must be produced is 24 hours.

Therefore, if a machine is said to have a refrigerative capacity of 50 tons, it means that the machine can produce 50 tons in 24 hours. One ton of refrigeration is equivalent to the abstraction of 200 B.Th.U. per minute. Ice-making capacity is, of course, always less than refrigerative capacity, since not only must the water be cooled to 32 deg. Fahr., but manufactured ice is generally far below 32 deg. because the temperature of the bath in which it is made is often in the neighbourhood of 12 deg. Fahr.

Hence we have the following equations :

Latent heat of 2000 pounds of ice . . . 288,000 B.Th.U.

In America the temperature of the water in the city main at midsummer may very well rise to 90 deg. Fahr. Thus we have

Sensible heat of water, 2000 times (90-32) . . . 116,000.

If the specific heat of ice be taken as 0.5 we also have
Sensible heat of ice, 200 times 0.5 (32-12) . . . 20,000,
giving a total of 466,400 when 10 per cent. is added for losses.

$\frac{466,000}{288,000}$ equals 1.6, and hence it is usual practice to call 1 ton of ice-making capacity equal to 1.6 tons of refrigeration.

Essential Features of Ice-Making Plant.—The essential features of an ice-making plant are shown in fig. 64.

These consist of a large tank, usually of sheet metal and well insulated on the outside, suitable evaporating coils and ice cans, together with their supporting framework. Frick standard freezing tank coils are of $1\frac{1}{4}$ inch selected ammonia pipe electrically welded into continuous

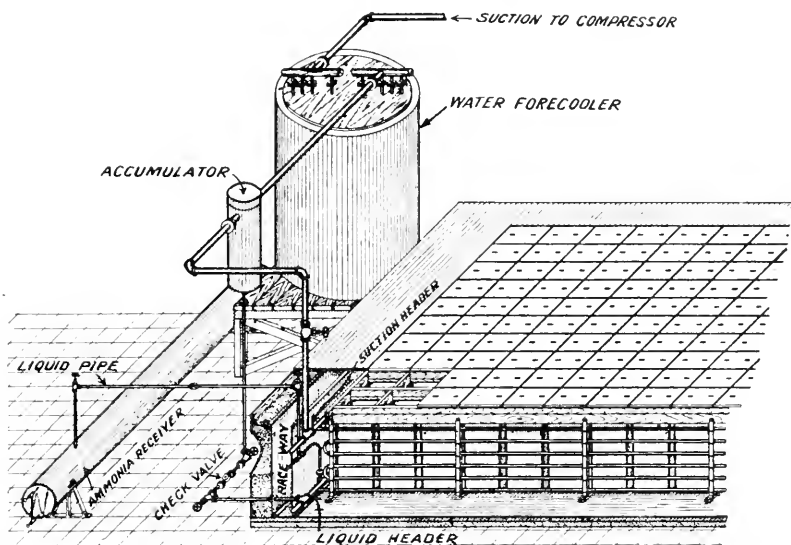


FIG. 64.—Ice-Making Plant.

lengths of 40 feet. Longer coils are made in two sections joined in the centre by standard four-bolt connections. The only other thread and gasket joints are those where these coils are joined to the ammonia feed and return headers. The tank is filled with brine above the level of the coils and circulation is caused by a brine agitator, a small impeller driven by a belt or electric motor. In modern plants what is known as the flooded system is used, that is, the ammonia coils are kept partially filled with liquid ammonia. This ensures a more rapid transfer of heat from the outside to the inside of the cooling coils. In order to prevent liquid ammonia from passing over to

the compressor along with dry gas a trap or accumulator (shown in the illustration) is provided. This trap is of sufficient size and depth to permit the liquid to drop to the bottom while the gas passes off at the top. The separated liquid automatically returns to the coils and there is a constantly repeated circulation of the ammonia. In order to prevent any gas from finding its way out of the trap by way of the return liquid line, a special check valve is placed at the bottom of the trap. This valve opens only when the head of the liquid in the trap is sufficient to prevent this happening.

When ordinary undistilled water is used for ice manufacture it is necessary, in order to make a clear merchantable product, to keep the water in lively agitation during the freezing period. This is done by means of compressed air, which is thoroughly cleaned, cooled and dried before it reaches the ice cans. The air is commonly led in by a central tube, hanging down inside the can. As the water freezes from the outside, a core is finally left containing the impurities.

This core may then be sucked out and refilled with fresh water from the storage tank.

The Production of Very Low Temperature.—Another aspect of the science of refrigeration of unusual interest is the production of very low temperature—temperatures sufficiently low to produce liquefaction of the so-called permanent gases, such as oxygen and nitrogen. Such processes find application in modern industry to the fixation of atmospheric nitrogen used in the manufacture of explosives, ammonia and fertilisers.

As already indicated, in order that a gas may be liquefied, it must be cooled below the critical temperature. This latter quantity has already been defined as that temperature above which no increase in pressure will cause liquefaction. In this connection reference may again be made to fig. 12 (p. 34), which shows the characteristic form of a series of isothermals for a fluid drawn between co-ordinates of pressure and volume.

In order therefore that a fluid, such as air, may be cooled to that point where condensation will commence

for any assigned pressure, we might use a train of refrigerating machines, the refrigerator of the first machine functioning as the condenser of the second, and so on. Thus we might employ for this purpose carbon dioxide in the first, ethylene in the next, and oxygen in the third. However, the two methods in common use are known respectively as the Linde and the Claude.

The first depends upon the Joule-Thomson cooling effect for its success, while the latter is a combination of the Joule-Thomson cooling effect and the lowering of the temperature which results when a gas expands doing work.

The Linde Process.—It may be remembered that, according to Joule's law, the internal energy of a perfect gas depends only upon the temperature. Since also the product of pressure and volume is constant when the temperature is constant, it follows that during any action involving constant total heat the temperature does not change. When, however, actual gases are throttled, by passing them from a region of higher to a region of lower pressure through a porous plug or other throttling device, a temperature difference is observed. This difference may be either positive or negative, the actual value depending upon the temperature and to some extent upon the pressure at which throttling takes place. Below the point of inversion, as it is termed, the effect is to cool the gas while on passing through this point no change in temperature would be observed. While the temperature of inversion varies, for most gases it is high, and the general effect is a cooling. Hydrogen is a notable exception to the general rule, the temperature of inversion being about -112 deg. Fahr., and hence the effect of throttling hydrogen at normal temperatures is to warm it.

Let, now, air, or other gas, be compressed in a high ratio to an upper pressure P_1 , the heat generated in the process being removed by cooling water. Then, if the compressed air be throttled back to its original pressure, a cooling amounting to about 0.5 deg. Fahr. for every 15 pounds per square inch pressure drop will result. If now, instead of allowing this air to escape at the lower

temperature and to take in heat from the surrounding atmosphere, we enclose the apparatus in a thermally insulated chamber in such a manner that the air cooled by throttling is warmed by the inflowing air and is thereby restored to its original temperature, we have a means of continuously extracting heat from the chamber and its contents.

For, let the total heat of the air entering under pressure P_1 and at temperature T_1 be I_{en} . Then the total heat of the air leaving at the lower pressure P_2 , but at the original temperature T_1 , must be greater than this, the measure of the difference being given by $C_p (T_1 - T_2)$, in which C_p is the mean specific heat at constant pressure for the range T_2 to T_1 .

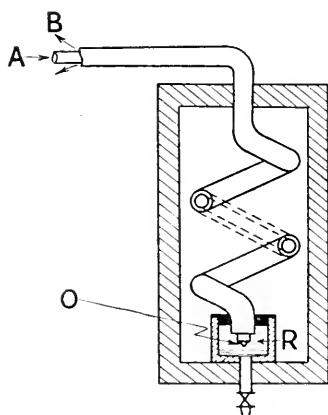


FIG. 65.—Apparatus necessary for production of low temperatures.

T_1 is the temperature before a throttling action without precooling, and T_2 is the temperature after a throttling action without precooling.

Further, the above statement will always be true, whatever be the actual conditions as to temperature inside the apparatus, so long as the air at exit is restored to T_1 .

Referring to fig. 65, which shows in diagrammatic form the necessary apparatus for carrying out this process, air under high pressure enters at A and flows through the inner coil to the orifice at O. Here it suffers a large pressure, but comparatively small temperature drop at constant total heat.

Thence it passes by way of the annular space between the inner and outer coils back to the atmosphere at B, being warmed in the process to the initial temperature T_1 . As the action proceeds and more and more heat is withdrawn with every pound of fluid which passes through

the coils, the temperature at the orifice will sink lower and lower and a temperature gradient will be established between the air at entrance and just before the passage of the throttle valve on the one hand, and between the air immediately after the passage and the exit on the other.

If the insulation of the chamber is such as to prevent thermal equilibrium being attained, the temperature will drop until condensation commences, when further fall will be arrested and liquid air will accumulate in the chamber R, whence it may be drawn off at will. The heat entering the apparatus is now equal to that leaving, and thus we have the following equations :

Let I_{en} be equal, as before, to the total heat per pound of the air entering the apparatus, reckoned from any convenient datum. Also let

I_{ex} be equal to the total heat of the air on exit and let x be the fraction of each pound liquefied.

Then $I_{en} = (1 - x) I_{ex} + x I_{liq}$

I_{liq} being the total heat of the liquid air.

Now, since $I_{ex} - I_{en} = \text{a constant, } Q$, say,

$$\begin{aligned}\therefore Q &= I_{ex} - (1 - x) I_{ex} - x I_{liq} \\ &= x (I_{ex} - I_{liq}) \\ &= x \{L + C_p (T_1 - T_o)\}\end{aligned}$$

in which T_o is the temperature at which the liquid air boils in the region of low pressure on the discharge side of the expansion valve, L is the latent heat of evaporation of the liquid, and C_p is the mean specific heat of the air between T_o and T_1 .

$$\text{Thus} \quad x = \frac{Q}{L + C_p (T_1 - T_o)}$$

from which the fraction which may be liquefied ideally may be determined when Q is known. In actual practice, of course, heat will leak into the machine, and, further, the temperature of the air leaving the outer coil would necessarily be lower than that of the entering air.

The fraction x , however, may be increased by increasing the pressure drop, and also by lowering the

initial temperature T_1 . This latter may be accomplished by arranging that the air circulate in a closed circuit, additional air being introduced only in order to make up the deficiency due to liquefaction. That is, the air on leaving instead of being permitted to pass to the atmosphere is led to the compressing pump, and so returned to the system.

Claude's Method.—In this process an expansion cylinder takes the place of the orifice in the Linde apparatus. Air under high pressure is allowed to expand after having its temperature lowered in a counter-current thermal interchanger similar in principle to the arrangement already described.

A part of the high-pressure air is diverted before entering the cylinder and passes into a condenser, cooled by the main mass of the air which has had its temperature lowered by expansion. The air in the inner tubes of the condenser, being at a much higher pressure than the cooling air, condenses and may then be drawn off from the receiving chamber placed at the lower end. Early attempts to reach very low temperature in this way were unsuccessful on account of lubrication difficulties. Claude, however, used certain hydrocarbons as lubricants until the action had started, when the liquid itself acted as the lubricant.

TOPICS FOR DISCUSSION

1. In gaseous explosions neither the temperatures nor the pressures expected on the assumption of constancy of the specific heat are realised. Give some explanation of this.

2. Tell how the specific heat of a gas may be determined experimentally.

3. Find an expression for the change in entropy per unit mass of a gas for which the specific heat at constant pressure is known, and the temperature and pressure limits are assigned.

4. Derive an equation relating the indicated mean pressure of the Diesel cycle, and the pressures and volumes at the corners of the diagram, γ the ratio of the specific heats being known.

5. Discuss the circumstances which limit the commercial utility of the gas-turbine.

6. Show how a temperature-entropy chart may be constructed for use with gases of varying characteristics. If non-rectangular co-ordinates are employed to represent the relation between temperature and entropy for a given working substance, will this have any effect upon the area enclosed by a reversible cycle?

7. Sketch the constant pressure cycle of the internal combustion turbine with isothermal compression, between co-ordinates of temperature and entropy. Indicate the areas representing turbine work ideally available, heat rejected, and work done by the compressor. Show how the casing temperature might be determined for any assumed turbine efficiency.

8. What is meant by regeneration? Indicate, by

reference to the $T\phi$ diagram of Question 7, the effect of regeneration upon the amount of heat which must be supplied from external sources.

9. Given the adiabatic heat drop, how would the 'spouting velocity' of the gases issuing from a turbine nozzle be determined?

10. Show that, the work done upon the air-screw remaining constant, the velocity of flight of an air-plane is inversely proportional to the cube root of the density of the atmosphere.

11. Find an expression for the ratio in which the atmospheric pressure falls, in terms of the elevation above the earth's surface, on the assumption that the temperature remains constant and equal to 32 deg. Fahr. Assuming a linear variation of temperature with elevation of 1 deg. per 333 feet, modify your expression to take account of the temperature change.

12. Trace the cycle of the vapour compression refrigerating machine employing wet compression (the working substance being dry at the end of compression) between co-ordinates of pressure and volume, and between co-ordinates of temperature and entropy. Show on the temperature-entropy diagram the influence of undercooling of the liquid below the temperature of condensation, and explain how the curve exhibiting the movement of the state point during the throttling action in the expansion valve may be plotted.

13. Describe the Linde process used in the manufacture of liquid air, and derive the expression

$$x = \frac{Q}{L + C_p (T_1 - T_o)}$$

WORKED EXAMPLES

1. The following particulars apply to a two-stage air compressor: Stroke, 10 inches; low pressure diameter, 11.5 inches; final air pressure, 350 pounds per square

inch ; intermediate pressure, 72 pounds per square inch ; temperature of the air leaving the intercooler, 95 deg. Fahr. If the volume of the air drawn into the compressor and measured at 15 pounds per square inch and 60·8 deg. Fahr. is 80 per cent. of the low-pressure swept cylinder volume, find the horse-power expended in compressing air when running at 350 revolutions per minute. Assume $PV^{1.3}$ equals a constant. [B.Sc. (Eng.) Lond.]

If the induction of the air into the cylinder be considered as a throttling process, then the total heat remains constant and, in the case of air, the temperature also is sensibly constant. During intake, therefore, the gas obeys Boyle's law, and the pressure within the cylinder of the compressor is

$$0.80 \times 15 = 12.0 \text{ pounds per square inch.}$$

V_1 , the swept volume of the low-pressure cylinder, is equal to

$$\frac{\pi d^2}{4} l = 0.6010 \text{ cubic foot.}$$

V_2 , the volume at the end of compression in the first stage, is given by

$$V_1 \left(\frac{P_1}{P_2} \right)^{1.3} = 0.1516.$$

The work done in the low-pressure stage may now be found from

$$\begin{aligned} & P_2 V_2 + \frac{P_2 V_2 - P_1 V_1}{n - 1} - P_1 V_1 \\ &= \frac{n - 1}{n} \left\{ P_2 V_2 - P_1 V_1 \right\} \\ &= \frac{1.3 \times 144}{0.3} \left\{ 72 \times 0.1516 - 12 \times 0.6010 \right\} \\ &= 2314 \text{ foot-pounds.} \end{aligned}$$

The temperature at the end of compression in the first stage must now be determined :

$$\begin{aligned} T_2 &= T_1 \frac{P_2 V_2}{P_1 V_1} \\ &= 520 \cdot 8 \times \frac{72 \times 0 \cdot 1516}{12 \times 0 \cdot 6010} \\ &= 788 \cdot 1^\circ \text{ Fahr. abs.} \end{aligned}$$

The temperature of the air leaving the intercooler
 $= 460 + 95 = 555^\circ \text{ Fahr. abs.}$

Hence V_3 , the volume of the air at the commencement of compression in the high-pressure cylinder, is equal to

$$\begin{aligned} V_3 &= V_2 \frac{T_3}{T_2} \\ &= 0 \cdot 1516 \times \frac{555}{788 \cdot 1} \\ &= 0 \cdot 1068 \text{ cubic foot.} \end{aligned}$$

Work done in second stage

$$= \int_{P_3}^{P_4} \frac{P_4}{P_3} V dP$$

Now $V = V_3 \left(\frac{P_3}{P} \right)^{1 \cdot 3}$

$$\begin{aligned} \therefore \text{work done} &= V_3 P_3^{1 \cdot 3} \int_{P_3}^{P_4} P^{-\frac{1}{1 \cdot 3}} dP \\ &= \frac{V_3 P_3^{1 \cdot 3} \left\{ P_4^{1 - \frac{1}{1 \cdot 3}} - P_3^{1 - \frac{1}{1 \cdot 3}} \right\}}{1 - \frac{1}{1 \cdot 3}} \\ &= \frac{0 \cdot 1068 (72 \times 144)^{\frac{1}{1 \cdot 3}} \left\{ (350 \times 144)^{3/13} - (72 \times 144)^{3/13} \right\}}{\frac{3}{1 \cdot 3}} \\ &= 2114 \text{ foot-pounds.} \end{aligned}$$

Hence the total work done per revolution is

$$2314 + 2114 = 4428 \text{ foot pounds.}$$

The horse-power expended in compressing the air is therefore

$$\frac{4428 \times 350}{33000} \\ = 46.97$$

2. Calculate the ideal efficiency of an internal combustion engine having a stroke of 16 inches, a piston diameter of 12 inches, and a clearance volume of 485 cubic inches. Find, also, the gas consumption per I.H.P. hour if the gas has a calorific value of 468 B.Th.U. per cubic foot and the efficiency ratio of the engine is 56 per cent. [B.Sc. (Eng.) Lond.]

The volume swept through by the piston is equal to

$$\frac{\pi d^2}{4} l \\ = 1810 \text{ cubic inches.}$$

Swept vol. + clearance vol. = 1810 + 485 = 2295 cu. in.

$$\text{Efficiency} = 1 - \left(\frac{1}{r}\right)^{\gamma-1} = 1 - \left(\frac{485}{2295}\right)^{0.4} = 46.3 \text{ per cent.}$$

$$\text{Actual efficiency} = 0.56 \times 0.463 = 25.93 \text{ per cent.}$$

$$\text{Gas consumption} = \frac{33000 \times 60}{778 \times 0.2593 \times 468} \\ = 20.98 \text{ cubic feet per I.H.P. hour.}$$

3. Calculate the stroke and diameter of a gas-engine which is to develop 15 horse-power at 300 revolutions per minute.

Assume a compression ratio of 4.5, adiabatic compression, a pressure at the end of explosion of 350 pounds per square inch, and an expansion index of 1.3. The stroke is to be 1.5 times the diameter. [B.Sc. (Eng.) Lond.]

$$\begin{aligned}
 P_m &= \frac{P_1 V_{cl} - P_2 V_e - P_4 V_{cl} - P_3 V_e}{n - 1} \frac{\gamma - 1}{V_e - V_{cl}} \\
 &= \frac{P_1 V_{cl} - P_1 \left(\frac{1}{r}\right)^n V_{cl} r - P_3 r^\gamma V_{cl} - P_3 r V_{cl}}{n - 1} \frac{\gamma - 1}{r V_{cl} - V_{cl}} \\
 &= \frac{P_1 (1 - r^{1-n})}{n - 1} - \frac{P_3 (r^\gamma - r)}{\gamma - 1} \\
 &= \frac{350 (1 - 0.6368)}{0.3} - \frac{14.7 \times 3.713}{0.4} \\
 &= \frac{3.5}{3.5} \\
 &= 82.1 \text{ pounds per square inch.} \\
 d &= \left\{ \frac{\text{H.P.} \times 33000 \times 12 \times 4}{p_m \times 1.5 \times \pi \times N} \right\}^{\frac{1}{3}} \\
 &= 7.425 \text{ inches.} \\
 l &= 11.137 \text{ inches.}
 \end{aligned}$$

4. Find an expression for the ideal efficiency of the Diesel engine cycle, and calculate this efficiency in case of an engine having a compression ratio of 13.5 and in which the fuel is cut off at 0.05 stroke. [B.Sc. (Eng.) Lond.]

The efficiency of the Diesel, as shown in the text, is

$$1 - \left(\frac{1}{r}\right)^{\gamma-1} \frac{r_1^\gamma - 1}{\gamma (r_1 - 1)}$$

Now

$$r = 13.5,$$

also

$$V_2 - V_1 = 0.05 (V_e - V_1)$$

$$\therefore V_2 - V_1 = 0.05 (13.5 V_1 - V_1)$$

$$V_2 = (0.05 \times 12.5) + V_1$$

$$\therefore \frac{V_2}{V_1} = 1.625$$

$$\begin{aligned}
 \text{Hence efficiency} &= 1 - \left(\frac{1}{13.5} \right)^{0.4} \times \frac{1 \cdot 625^{1.4} - 1}{1.4 \times 0.625} \\
 &= 1 - (0.3530 \times 1.112) \\
 &= 60.75 \text{ per cent.}
 \end{aligned}$$

5. Calculate the bore and stroke of a Diesel engine which will develop 25 indicated horse-power at 300 revs. per min. and have compression ratio 14 and cut-off at 4.5 per cent. of the stroke. Take the stroke as 1.25 times the diameter, and assume for the compression and expansion curves indices of 1.4 and 1.3 respectively, and a pressure at the beginning of compression of 13.5 pounds per square inch. [B.Sc. (Eng.) Lond.]

Using the notation already employed in the text,

$$P_m = \frac{P_1(V_2 - V_1) + \frac{P_1 V_2 - P_3 V_c}{n - 1} - \frac{P_1 V_1 - P_4 V_c}{\gamma - 1}}{V_c - V_1}$$

$$\text{Now } \frac{V_c}{V_1} = r \text{ and } \frac{V_2}{V_1} = r_1$$

$$P_1 = P_4 r^\gamma \text{ and } P_3 = P_1 \left(\frac{V_2}{V_c} \right)^n = P_4 r^\gamma \left(\frac{r_1}{r} \right)^n$$

$$P_m =$$

$$\frac{P_4 r^\gamma (V_1 r_1 - V_1) + \frac{P_4 r^\gamma V_1 r_1 - P_4 r^\gamma \left(\frac{r_1}{r} \right)^n V_1 r}{n - 1} - \frac{P_4 r^\gamma V_1 - P_4 V_1 r}{\gamma - 1}}{V_1 r - V_1}$$

$$\begin{aligned}
 &= \frac{P_4 r^\gamma V_1 (r_1 - 1) + \frac{P_4 V_1 \left[r^\gamma r_1 - r^{\gamma+1} \left(\frac{r_1}{r} \right)^n \right]}{n - 1} - \frac{P_4 V_1 (r^\gamma - r)}{\gamma - 1}}{V_1 (r - 1)}
 \end{aligned}$$

$$= \frac{P_4 \left\{ r^\gamma (r_1 - 1) + \frac{r^\gamma r_1 - r^{\gamma+1} \left(\frac{r_1}{r} \right)^n}{n - 1} - \frac{r^\gamma - r}{\gamma - 1} \right\}}{r - 1}$$

$$= \frac{13.5 \left\{ 14^{1.4(0.585)} + \frac{14^{1.4} \cdot 1.585 - 14^{2.4} \left(\frac{1.585}{14} \right)^{1.3}}{0.3} - \frac{14^{1.4} - 14}{0.4} \right\}}{14 - 1}$$

$p_m = 61.95$ pounds per square inch.

If this engine be taken as a four-stroke single-acting Diesel, we have

$$\begin{aligned} d &= \left(\frac{4 \times 33000 \times \text{H.P.} \times 12}{P_m \times 1.25 \times \frac{1}{2} \times N \times \pi} \right)^{\frac{1}{3}} \\ &= 10.28 \text{ inches.} \\ l &= 12.85 \text{ inches.} \end{aligned}$$

6. Explain in what respect the cycle of operations in a vapour compression refrigerating plant differs from the reversed Carnot cycle.

The temperature range in an ammonia plant is from 68 deg. Fahr. to 14 deg. Fahr. Find the coefficient of performance for the corresponding reversed Carnot cycle and also for the ammonia plant if after compression the ammonia gas is 0.95 dry.

What improvement in the coefficient of performance would be obtained if the plant were so managed that the temperature of the gas after compression were 95 deg. Fahr., the pressure range remaining as before?

The values of the entropy for the liquid and dry vapour at 68 deg. Fahr. are 0.2494 and 1.2170, and at 14 deg. Fahr. they are 0.1300 and 1.3081 respectively, and the specific heat of the vapour is 0.670. [B.Sc. (Eng.) Lond.]

As indicated in the text, the adiabatic expansion of the reversed Carnot cycle is replaced in the actual refrigeration cycle by an expansion at constant total heat. This action takes place in the valves which control the flow of ammonia liquor to the cooling coils.

Fig. 20 (p. 43) shows that the influence of this change is to reduce the refrigerative effect for the same expenditure of work. Thus the coefficient of performance is less.

Coefficient of performance for the reversed Carnot

$$\text{cycle} = \frac{T_2}{T_1 - T_2} = \frac{474}{54} = 8.778.$$

Referring to fig. 20, the coefficient of performance for the actual cycle = $\frac{\text{area (iii) (iv) (vi) (vii)}}{\text{area (i) (ii) (iii) (v)}} = \frac{I_{\text{(iii)}} - I_{\text{(iv)}}}{I_{\text{(ii)}} - I_{\text{(iii)}}}$ very approximately.

If the curve (i) (v) may be regarded as a straight line (which is very approximately true), and the $\frac{PV_{liq}}{J}$ term be neglected,

$$\begin{aligned} I_{\text{(ii)}} - I_{\text{(v)}} &= (\phi_{\text{(i)}} - \phi_{\text{(v)}}) \frac{T_{\text{(i)}} + T_{\text{(v)}}}{2} + (\phi_{\text{(ii)}} - \phi_{\text{(i)}}) T_{\text{(i)}} \\ &= (0.2494 - 0.1300) \frac{528 + 474}{2} \\ &\quad + 0.95 (1.2170 - 0.2494) 528 \\ &= 0.1194 \times 501 + 0.95 \times 0.9676 \times 528 \\ &= 59.82 + 485.3 \\ &= 545.12 \text{ B.Th.U.} \end{aligned}$$

$$\begin{aligned} I_{\text{(iii)}} - I_{\text{(v)}} &= (\phi_{\text{(iii)}} - \phi_{\text{(v)}}) T_{\text{(v)}} = (\phi_{\text{(ii)}} - \phi_{\text{(v)}}) T_{\text{(v)}} \\ &= (0.95 \times 0.9676 + 0.1194) 474 \\ &= 492.3 \text{ B.Th.U.} \end{aligned}$$

$$\begin{aligned} \therefore I_{\text{(ii)}} - I_{\text{(iii)}} &= 545.12 - 492.3 \\ &= 52.82 \text{ B.Th.U.} \end{aligned}$$

(Note that if the quantity $\frac{PV_{liq}}{J}$ were taken account of, $I_{\text{(v)}}$ would not be the same for both.)

Since the total heat at (i) equals the total heat at (iv)

$$\begin{aligned} I_{\text{(iii)}} - I_{\text{(iv)}} &= I_{\text{(iii)}} - I_{\text{(i)}} \\ &= 492.3 - 59.82 \\ &= 432.48 \text{ B.Th.U} \end{aligned}$$

$$\text{Coefficient of performance} = \frac{432.48}{52.82} = \underline{8.19}.$$

Required next to determine what improvement in the coefficient takes place when the temperature at the end of compression is 95 deg. Fahr.

$$\text{Coefficient of performance} = \frac{I_{(iii)'} - I_{(iv)}}{I_{(ii)} - I_{(iii)}}$$

$$I_{(iii)'} - I_{(v)} = (\phi_{(i)} - \phi_{(v)}) \frac{T_{(i)} + T_{(v)}}{2} + (\phi_{(ii)} - \phi_{(i)}) T_{(i)} \\ + (\phi_{(ii)'} - \phi_{(ii)}) \frac{T_{(ii)'} + T_{(ii)}}{2}$$

$$= 59.82 + 0.9676 \times 528 + C_p \log_e \frac{T_{(ii)'}}{T_{(ii)}} \times \frac{555 + 528}{2}$$

$$= 570.32 + 0.67 \times 2.303 \times \log_{10} \frac{555}{528} \times 541.5$$

$$= 570.32 + 18.13$$

$$= 588.45 \text{ B.Th.U.}$$

$$I_{(iii)'} - I_{(v)} = (\phi_{(iii)'} - \phi_{(v)}) T_{(v)} = (\phi_{(ii)'} - \phi_{(v)}) T_{(v)} \\ = (0.1194 + 0.9676 + 0.0335) 474 \\ = 1.1205 \times 474 \\ = 531.1 \text{ B.Th.U.}$$

$$I_{(ii)'} - I_{(iii)'} = 588.45 - 531.1 = 57.35 \text{ B.Th.U.}$$

$$I_{(iii)'} - I_{(iv)} = I_{(iii)'} - I_{(i)} \\ = 531.1 - 59.82 \\ = 471.28 \text{ B.Th.U.}$$

$$\text{Coefficient of performance} = \frac{471.28}{57.35} = \underline{\underline{8.22}}$$

These calculations may be checked by reference to the Mollier chart of pressure and total heat.

EXAMPLES FOR PRACTICE

1. The compression ratio in both cylinders of a two-stage air compressor is to be the same, and the air is cooled at constant pressure in the intercooler to 77 deg. Fahr.

If air is drawn into the compressor at 14 pounds per square inch and 50 deg. Fahr. and is to be delivered at

500 pounds per square inch, find the ratio of compression in the cylinders and the work done in compressing a pound of air.

Assume for compression curves that $PV^{1.35}$ equals a constant. [B.Sc. (Eng.) Lond.]

2. A single-acting two-stage compressor compressing air for a Diesel engine has piston diameters 4 inches and $1\frac{1}{2}$ inches and a stroke in each cylinder of 3 inches. Air is drawn into the low-pressure cylinder at atmospheric conditions of 14.7 pounds per square inch and 59 deg. Fahr., and is cooled in the intercooler to 59 deg. Fahr. From the high-pressure cylinder it is delivered into a storage bottle having a pressure of 600 pounds per square inch. Neglecting clearance volumes, determine the pressure in the intercooler when the compressor is running steadily and the intercooler pressure is steady, and find the horse-power used in compressing the air at 350 revs. per min. [B.Sc. (Eng.) Lond.]

3. Calculate the diameter and stroke of a gas-engine which can develop 25 brake horse-power at 300 revs. per min., assuming a mechanical efficiency of 80 per cent., a mean effective pressure of 85 pounds per square inch, and a ratio of stroke to diameter of 1.5.

If the clearance volume is one-quarter of the swept cylinder volume and the actual thermal efficiency (I.H.P. basis) is 55 per cent. of the ideal efficiency, what will be the consumption per brake horse-power hour of gas having a calorific value of 495 B.Th.U. per cubic foot? [B.Sc. (Eng.) Lond.]

4. The cylinders of a petrol motor have stroke and bore 120 and 90 mm. respectively and a compression ratio of 4.4. The inlet valve is shut at 0.05 stroke and the pressure and temperature of the charge is then 13.5 pounds per square inch and 203 deg. Fahr. The exhaust valve opens at 0.85 stroke, and the pressure at this point is indicated as 51 pounds per square inch. Assuming a molecular contraction of 2 per cent., find the temperature of the cylinder contents at the opening of the exhaust valve, and estimate the heat discharged in the exhaust per minute if the motor has four cylinders and is running at

1000 revs. per min. Take the mean specific heat of the exhaust gases as 0.24. [B.Sc. (Eng.) Lond.]

5. A gas engine uses producer gas which has a volumetric percentage analysis of CO, 11.0; H_2 , 29.0; CH_4 , 1.8; CO_2 , 16.1; N_2 , 42.1. Calculate the volume of air required to completely burn a cubic foot of this gas.

If the air supplied to the engine be 50 per cent. in excess of this quantity, what is the calorific value of a cubic foot of cylinder mixture? Calorific values of CO, H_2 , and CH_4 are 342, 291, and 963 B.Th.U. per cubic foot respectively. What is the gas consumption per horse-power hour if the engine efficiency is 23.6 per cent.? [B.Sc. (Eng.) Lond.]

6. The following results were obtained during a test of a gas-engine loaded by a friction brake :

Cylinder diameter, 8 ins.; stroke, 17 ins.; dead weight, 165 pounds; spring balance reading, 23.8 pounds; brake wheel diameter, 5 feet; revs. per min., 215; explosions per min., 98; mean effective pressure of indicator card, 82 pounds per square inch; gas per min., 7.16 cubic feet at 29.9 inches of mercury and 58.6 deg. Fahr.; cooling water per min., 37.7 pounds raised 47.6 deg. Fahr.; calorific value of gas, 495 B.Th.U. per cubic foot, measured at normal temperature and pressure.

Calculate the indicated and brake horse-powers of the engine, and find the mechanical and thermal efficiencies.

Draw up a heat balance-sheet for the engine per minute. [B.Sc. (Eng.) Lond.]

7. The two series of trials from which the following observations were taken were made on the same gas-engine and under exactly the same conditions, except that in series A the engine was controlled by a hit-and-miss governor and in series B the control was a throttle control effected by regulation of the gas cock.

Plot on a brake horse-power base curves showing gas consumption per minute and also gas consumption per brake horse-power hour for these two series of trials.

Comment on these curves, and especially give reasons for the differences in efficiency occasioned by the two methods of control.

Series A

Brake horse-power	. 6.20	5.33	4.07	2.79	1.43	0
Gas per min., cu. ft.	3.94	3.65	3.17	2.71	2.28	1.90

Series B

Brake horse-power	. 6.13	5.17	4.01	2.69	1.28	0
Gas per min., cu. ft.	3.90	3.58	3.25	3.00	2.74	2.51

[B.Sc. (Eng.) Lond.]

8. The pressures on the compression curve of a Diesel engine diagram are found to be at $\frac{1}{4}$ stroke 23.4, and at $\frac{7}{8}$ stroke 161 pounds per square inch. Estimate the compression ratio of this engine. Calculate the ideal efficiency of this engine if cut-off takes place at $\frac{1}{15}$ stroke. [B.Sc. (Eng.) Lond.]

9. In a trial on a four-stroke single-acting Diesel engine having a piston diameter 24.5 inches and stroke 39 inches, the following observations were made: speed, 125 revs. per min.; mean effective pressure, 92 pounds per square inch, shaft torque, 8670 pounds-feet; horse-power used to compress blast air, 15; fuel used per minute, 1.45 pound of oil having a calorific value of 19,350 B.Th.U. per pound and containing 85 per cent. of carbon and 12 per cent. of hydrogen by weight; cooling water, 148 pounds per minute, raised 68 deg. Fahr. The weight of exhaust gas was 29.1 pounds per pound of fuel burnt. The exhaust gas left the engine at a temperature of 603 deg. Fahr. and its specific heat may be taken as 0.25.

Make out a heat balance-sheet for the engine, showing heat quantities in B.Th.U. per minute. [B.Sc. (Eng.) Lond.]

10. In a Bell-Coleman refrigerating plant air is drawn into the cylinder of the compressor at atmospheric pressure of 15 pounds per square inch and temperature of 23 deg. Fahr., and it is compressed adiabatically to 75 pounds per square inch, at which pressure it is cooled to 59 deg. Fahr. It is then expanded in an expansion cylinder to atmospheric pressure and discharged into the refrigerating chamber.

If the law for expansion is $PV^{1.2} = \text{a constant}$, find the work done on the air per pound and the coefficient of performance of the refrigerating plant. The specific heat of air at constant pressure is 0.238 . [B.Sc. (Eng.) Lond.]

11. Determine the theoretical coefficient of performance of a CO_2 refrigerating plant working over a pressure range of 826 to 350 pounds per square inch if the temperature of the gas after compression is 95°Fahr. , given that at 826 pounds per square inch pressure the entropy of the liquid is 0.045 , the temperature of evaporation 68°Fahr. , and the latent heat 66.4 B.Th.U. ; and that at 350 pounds per square inch pressure these values are, respectively, -0.025 , 7.9°Fahr. , and 114.1 B.Th.U. The specific heat of CO_2 gas is 0.56 . [B.Sc. (Eng.) Lond.]

12. Explain why the theoretical coefficient of performance of an ammonia vapour compression refrigeration plant is greater than that of a carbon-dioxide plant working over the same range of temperature.

Compare the theoretical coefficients of performance of such plants working over a range of 68°Fahr. to -4°Fahr. and arranged so that the vapour is just dry and saturated at the end of compression.

Ammonia.				Carbon Dioxide.		
t .	p	ϕ_{liq}	$\frac{L}{T}$	p	ϕ_{liq}	$\frac{L}{T}$
68°Fahr.	124.3	0.2494	0.9676	826	0.045	0.171
-4°Fahr.	27.6	0.0880	1.2553	289	-0.036	0.226

Vol. of 1 pound of liquid ammonia, 0.0256 cubic feet; vol. of 1 pound of liquid carbon dioxide, 0.0176 . [B.Sc. (Eng.) Lond.]

(In estimating total heat quantities for the purposes of this problem it is intended that the PV_{liq}/J term be taken into account. Heat interchanges may be estimated from the temperature-entropy diagram on the assumption that the liquid line is straight.)

13. In an ammonia refrigeration plant in which the compressor was driven directly by a steam engine, the following observations were made: Horse-power of steam cylinder, 15.5 ; horse-power of compressor cylinder,

13.7; heat abstracted from the brine per hour, 172,000 B.Th.U.; temperature of the brine, 14 deg. Fahr.; temperature of cooling water, 68 deg. Fahr.; steam supply to cylinder, 20.7 pounds per I.H.P. hour; heat per pound of steam from feed temperature, 1089 B.Th.U.

Calculate :

(1) The theoretical coefficient of performance for the ammonia cycle, wet compression (that is, the working substance is just dry and saturated at the end of the adiabatic compression) over the range 68 deg. Fahr. to 14 deg. Fahr.;

(2) The actual coefficient of performance of the plant;

(3) The ratio of the heat abstracted by the plant to the heat supplied to the plant.

The specific heat of liquid ammonia for this range may be taken as 1.11, and the latent heats at 68 deg. Fahr. and 14 deg. Fahr. are 510.5 and 557.9 B.Th.U. respectively. [B.Sc. (Eng.) Lond.]

TABLE OF THE
THERMODYNAMIC PROPERTIES
OF SATURATED STEAM

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapour.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
p	t	l_{liq}	L	$l_s = l_{liq} + L$	l_{int}	l_{ext}	ϕ_{liq}	$\frac{L}{T}$	$\phi_{liq} + \frac{L}{T}$	v_s	w_s
† 0.1	35.03	3.05	1071.7	1074.7	1017.3	54.4	0.0002	2.1666	2.1728	2935.0	0.000340
† 0.2	53.15	21.23	1061.6	1082.8	1005.2	50.5	0.0423	2.0704	2.1127	1524.0	0.000656
† 0.3	64.49	32.57	1055.3	1087.9	997.7	57.6	0.0640	2.0135	2.0775	1041.0	0.000961
0.4	72.91	40.95	1050.6	1091.6	992.4	58.5	0.0860	1.9730	2.0530	794.0	0.001259
0.5	79.68	47.71	1047.0	1094.6	987.6	59.3	0.0926	1.9413	2.0332	642.0	0.001555
0.6	85.32	53.34	1043.8	1097.1	983.9	59.9	0.1029	1.9155	2.0184	541.0	0.001850
0.7	90.18	58.18	1041.1	1099.3	980.7	60.4	0.1117	1.8936	2.0053	467.0	0.002143
0.8	94.46	62.45	1038.7	1101.2	977.8	61.0	0.1195	1.8747	1.9942	412.0	0.002431
0.9	98.33	66.31	1036.6	1102.9	975.2	61.4	0.1265	1.8578	1.9843	367.9	0.002719
1	101.83	69.8	1034.6	1104.4	972.9	61.7	0.1327	1.8427	1.9754	333.0	0.003000
2	126.15	94.0	1021.0	1115.0	950.7	64.3	0.1749	1.7431	1.9180	173.5	0.00576
3	141.52	109.4	1012.3	1121.6	940.4	65.8	0.2068	1.6840	1.8848	118.5	0.00845
4	153.01	120.9	1005.7	1126.5	938.6	67.0	0.2198	1.6416	1.8614	90.5	0.01107
5	162.28	130.1	1000.3	1130.5	932.4	68.0	0.2348	1.6084	1.8432	73.33	0.01364
6	170.00	137.9	995.8	1133.7	927.0	68.8	0.2571	1.5814	1.8285	61.89	0.01616
7	176.85	144.7	991.8	1136.5	922.4	69.4	0.2579	1.5582	1.8161	53.56	0.01867
8	182.86	150.8	988.2	1139.0	918.2	70.0	0.2673	1.5380	1.8053	47.27	0.02115
9	188.27	156.2	985.0	1141.1	914.4	70.6	0.2756	1.5202	1.7958	42.36	0.02361
10	193.22	161.1	982.0	1143.1	910.9	71.1	0.2832	1.5042	1.7847	38.38	0.02606
11	197.75	165.7	979.2	1144.9	907.8	71.5	0.2902	1.4895	1.7797	35.10	0.02849
12	201.96	169.9	976.6	1146.5	904.8	71.8	0.2967	1.4760	1.7727	32.36	0.03090

* Courtesy of the Publishers, Longmans, Green & Co., Ltd. † Interpolated.

PROPERTIES OF SATURATED STEAM—Continued.

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid, l_{liq}	Heat of Vaporization, L	Total Heat, $l_s = l_{liq} + L$	Heat Equivalent of Internal Work, l_{int}	Heat Equivalent of External Work, l_{ext}	Entropy of the Liquid, ϕ_{liq}	Entropy of the Vapour, $\frac{L}{T}$	Total Entropy, $\phi_{liq} + \frac{L}{T}$	Specific Volume, V_s	Density Weight per Cubic Foot, Pounds.
p	t	l_{liq}	L	$l_s = l_{liq} + L$	l_{int}	l_{ext}	ϕ_{liq}	$\frac{L}{T}$	$\phi_{liq} + \frac{L}{T}$	V_s	W_s
13	205.87	173.8	974.2	1148.0	902.0	72.2	0.3025	1.4639	1.7664	30.03	0.03330
14	209.55	177.5	971.9	1149.4	899.3	72.6	0.3081	1.4523	1.7604	28.02	0.03569
14.7	212.00	180.0	970.4	1150.4	897.6	72.9	0.3118	1.4447	1.7565	26.79	0.03732
15	213.0	181.0	969.7	1150.7	896.8	72.9	0.3133	1.4416	1.7549	26.27	0.03896
20	228.0	196.1	960.0	1156.2	885.8	74.3	0.3355	1.3965	1.7320	20.08	0.04980
25	240.1	208.4	952.0	1160.4	870.8	75.3	0.3532	1.3604	1.7136	16.30	0.0614
30	250.3	218.8	945.1	1163.9	860.0	76.2	0.3680	1.3311	1.6991	13.74	0.0728
35	259.3	227.9	938.9	1166.8	852.1	76.9	0.3808	1.3060	1.6868	11.89	0.0841
40	267.3	236.1	933.3	1169.4	855.9	77.6	0.3920	1.2841	1.6761	10.49	0.0953
45	274.5	243.4	928.2	1171.6	850.3	78.1	0.4021	1.2644	1.6665	9.39	0.1065
50	281.0	250.1	923.5	1173.6	845.0	78.6	0.4113	1.2468	1.6581	8.51	0.1175
55	287.1	256.3	919.0	1175.4	840.2	78.9	0.4199	1.2309	1.6505	7.78	0.1285
60	292.7	262.1	914.9	1177.0	835.6	79.7	0.4272	1.2160	1.6432	7.17	0.1394
65	298.0	265.7	911.0	1178.5	831.4	79.8	0.4344	1.2034	1.6368	6.65	0.1503
70	302.9	272.6	907.2	1179.8	827.3	80.1	0.4411	1.1896	1.6307	6.20	0.1612
75	307.6	277.4	903.7	1181.8	823.5	80.5	0.4474	1.1778	1.6252	5.81	0.1721
80	312.0	282.0	900.3	1182.3	819.8	80.7	0.4535	1.1665	1.6200	5.47	0.1829
85	316.3	286.3	897.1	1183.4	816.3	81.0	0.4590	1.1561	1.6151	5.16	0.1937
90	320.3	290.5	893.9	1184.4	813.0	81.2	0.4644	1.1461	1.6105	4.89	0.2044
95	324.0	294.5	890.9	1185.4	809.7	81.5	0.4694	1.1367	1.6061	4.65	0.2151
100	327.8	298.3	888.0	1186.3	806.6	81.7	0.4743	1.1277	1.6020	4.429	0.2258

105	331.4	302.0	885.2	1187.2	803.6	81.9	0.4789	1.1191	1.5980	4.230	0.2365
110	334.8	305.5	882.5	1188.0	800.7	82.1	0.4834	1.1108	1.5942	4.047	0.2472
115	338.4	309.0	879.8	1188.8	797.9	82.3	0.4877	1.1030	1.5907	3.880	0.2577
120	341.3	312.3	877.2	1189.6	795.2	82.5	0.4919	1.0954	1.5873	3.726	0.2683
125	344.4	315.5	874.7	1190.3	792.6	82.6	0.4959	1.0880	1.5839	3.583	0.2791
130	347.4	318.6	872.3	1191.0	790.0	82.8	0.4998	1.0809	1.5807	3.452	0.2897
135	350.3	321.7	869.9	1191.6	787.5	82.9	0.5035	1.0742	1.5777	3.331	0.3002
140	353.1	324.6	867.6	1192.2	785.0	83.0	0.5072	1.0675	1.5747	3.219	0.3107
145	355.8	327.4	865.4	1192.8	782.7	83.2	0.5107	1.0612	1.5719	3.112	0.3213
150	358.5	330.2	863.2	1193.4	780.4	83.3	0.5142	1.0550	1.5692	3.012	0.3320
155	361.0	332.9	861.0	1194.0	778.1	83.5	0.5175	1.0489	1.5664	2.920	0.3425
160	363.6	335.6	858.8	1194.5	775.8	83.6	0.5208	1.0431	1.5639	2.834	0.3529
165	366.0	338.2	856.8	1195.0	773.6	83.7	0.5239	1.0376	1.5615	2.753	0.3633
170	368.5	340.7	854.7	1195.4	771.5	83.8	0.5269	1.0321	1.5590	2.675	0.3738
175	370.8	343.2	852.7	1195.9	769.4	83.9	0.5299	1.0268	1.5567	2.602	0.3843
180	373.1	345.6	850.8	1196.4	767.4	84.0	0.5328	1.0215	1.5543	2.533	0.3948
185	375.4	348.0	848.8	1196.8	765.4	84.1	0.5359	1.0164	1.5520	2.468	0.4052
190	377.6	350.4	846.9	1197.3	763.4	84.2	0.5384	1.0114	1.5498	2.406	0.4157
195	379.8	352.7	845.0	1197.7	761.4	84.3	0.5410	1.0066	1.5476	2.346	0.4262
200	381.9	354.9	843.2	1198.1	759.5	84.4	0.5437	1.0019	1.5456	2.290	0.437
205	384.0	357.1	841.4	1198.5	757.6	84.5	0.5463	0.9973	1.5436	2.237	0.447
210	386.0	359.2	839.6	1198.6	755.8	84.5	0.5488	0.9928	1.5416	2.187	0.457
215	388.0	361.4	837.9	1199.2	754.0	84.6	0.5513	0.9885	1.5398	2.138	0.468
220	389.9	363.4	836.2	1199.6	752.3	84.7	0.5538	0.9841	1.5379	2.091	0.478
225	391.9	365.5	834.4	1199.9	750.5	84.7	0.5562	0.9799	1.5361	2.046	0.489
230	393.8	367.5	832.8	1200.2	748.8	84.8	0.5586	0.9758	1.5341	2.004	0.499
240	397.4	371.4	829.5	1200.9	745.4	85.0	0.5633	0.9676	1.5309	1.924	0.520
250	401.1	375.2	826.3	1201.5	742.0	85.1	0.5676	0.9600	1.5276	1.850	0.541
275	409.5	384.2	818.6	1202.8	734.2	85.3	0.5780	0.9419	1.5199	1.686	0.593
300	417.5	392.7	811.3	1204.1	726.8	85.6	0.5878	0.9251	1.5129	1.551	0.645

Absolute Pressure Pounds per Square Inch.	Tempera- ture, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapour.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
<i>p</i>	<i>t</i>	<i>i_{liq}</i>	<i>L</i>	<i>i_s</i> = <i>i_{liq}</i> + <i>L</i>	<i>i_{int}</i>	<i>i_{ext}</i>	<i>φ_{liq}</i>	$\frac{L}{T}$	$\phi_{liq} + \frac{L}{T}$	<i>V_s</i>	<i>W_s</i>
325	424.9	400.4	801.4	1201.8	717.0	84.7	0.5953	0.9065	1.5018	1.428	0.700
350	431.9	408.0	794.5	1202.5	709.7		0.6036	0.8912	1.4949	1.327	0.753
375	438.5	415.1	787.5	1202.6	702.7		0.6115	0.8768	1.4884	1.239	0.807
400	444.8	422.0	780.6	1202.5	695.9	84.6	0.6190	0.8631	1.4821	1.162	0.860
425	450.7	428.5	773.9	1202.4	689.4		0.6261	0.8501	1.4762	1.121	0.914
450	456.5	434.8	767.4	1202.2	683.1		0.6329	0.8377	1.4706	1.033	0.968
500	467.2	446.6	755.0	1201.7	670.9	84.0	0.6455	0.8146	1.4601	0.928	1.077
550	477.2	458.1	743.3	1200.8	665.2		0.6571	0.7934	1.4506	0.843	1.132
600	486.5	468.0	731.8	1199.8	648.5	83.0	0.6679	0.7735	1.4414	0.770	1.30
650	495.2	477.8	720.9	1198.7	638.0		0.6780	0.7550	1.4330	0.708	1.41
700	503.4	487.1	710.3	1197.4	627.9	82.3	0.6874	0.7376	1.4250	0.656	1.52
750	511.1	495.9	700.0	1195.9	618.2		0.6963	0.7212	1.4175	0.610	1.64
800	518.5	504.3	690.1	1194.4	608.8	81.2	0.7048	0.7056	1.4104	0.570	1.76
850	525.5	512.5	680.4	1192.8	599.7		0.7128	0.6907	1.4035	0.534	1.87
900	532.3	520.3	670.8	1191.9	590.8	80.1	0.7205	0.6764	1.3969	0.502	1.99
1000	544.9	535.2	652.4	1187.6	573.6	78.6	0.7349	0.6496	1.3845	0.447	2.24
1100	556.6	549.1	634.7	1183.8	557.3	77.6	0.7482	0.6247	1.3729	0.403	2.48
1200	567.7	562.3	617.6	1179.7	541.8	75.6	0.7607	0.6015	1.3622	0.364	2.74
1500	596.4	599.0	566.6	1167.0	495.0	73.5	0.7970	0.5360	0.1334	0.281	3.56
2000	635.9	657.0	480.0	1138.0	418.0	62.0	..	0.4390	..	0.194	5.18
3200	700.3	921.0	0.0	921.0	0.0	0.0	0.0	0.048	20.90

TABLE OF THE THERMODYNAMIC
PROPERTIES OF SUPERHEATED
STEAM

APPLIED TO ENGINEERING

PROPERTIES OF SUPERHEATED STEAM.

Reproduced by permission from Marks and Davis' 'Steam Tables and Diagrams.'

(Copyright, 1909, by Longmans, Green & Co.)

Pressure, Pounds Absolute.		Satur- ated Steam.	Degrees of Superheat.						Pressure, Pounds Absolute.	
			50	100	150	200	250	300		
5	<i>t</i>	162.3	212.3	262.3	312.3	362.3	412.3	462.3	<i>t</i>	
	V	73.3	79.7	85.7	91.8	97.8	103.8	109.8	V	5
	I	1130.5	1153.5	1176.4	1199.5	1222.5	1245.6	1268.7	I	
10	<i>t</i>	193.2	243.2	293.2	343.2	393.2	443.2	493.2	<i>t</i>	
	V	38.4	41.5	44.6	47.7	50.7	53.7	56.7	V	10
	I	1143.1	1166.3	1189.5	1212.7	1236.0	1259.3	1282.5	I	
15	<i>t</i>	213.0	263.0	313.0	363.0	413.0	463.0	513.0	<i>t</i>	
	V	26.27	28.40	30.46	32.50	34.53	36.56	38.58	V	15
	I	1150.7	1174.2	1197.6	1221.0	1244.4	1267.7	1291.1	I	
20	<i>t</i>	228.0	278.0	328.0	378.0	428.0	478.0	528.0	<i>t</i>	
	V	20.08	21.69	23.25	24.80	26.33	27.85	29.37	V	20
	I	1156.2	1179.9	1203.5	1227.1	1250.6	1274.1	1297.6	I	
25	<i>t</i>	240.1	290.1	340.1	390.1	440.1	490.1	540.1	<i>t</i>	
	V	16.30	17.60	18.86	20.10	21.32	22.55	23.77	V	25
	I	1160.4	1184.4	1208.2	1231.9	1255.6	1279.2	1302.8	I	
30	<i>t</i>	250.4	300.4	350.4	400.4	450.4	500.4	550.4	<i>t</i>	
	V	13.74	14.83	15.89	16.93	17.97	18.99	20.00	V	30
	I	1163.9	1188.1	1212.1	1236.0	1259.7	1283.4	1307.1	I	
35	<i>t</i>	259.3	309.3	359.3	409.3	459.3	509.3	559.3	<i>t</i>	
	V	11.89	12.85	13.75	14.65	15.54	16.42	17.30	V	35
	I	1166.8	1191.3	1215.4	1239.4	1263.3	1287.1	1310.8	I	
40	<i>t</i>	267.3	317.3	367.2	417.3	467.3	517.3	567.3	<i>t</i>	
	V	10.49	11.33	12.13	12.93	13.70	14.48	15.25	V	40
	I	1169.4	1194.0	1218.4	1242.4	1266.4	1290.3	1314.1	I	
45	<i>t</i>	274.5	324.5	374.5	424.5	474.5	524.5	574.5	<i>t</i>	
	V	9.39	10.14	10.86	11.57	12.27	12.96	13.65	V	45
	I	1171.6	1196.6	1221.0	1245.2	1269.3	1293.2	1317.0	I	
50	<i>t</i>	281.0	331.0	381.0	431.0	481.0	531.0	581.0	<i>t</i>	
	V	8.51	9.19	9.84	10.48	11.11	11.74	12.36	V	50
	I	1173.6	1198.8	1223.4	1247.7	1271.8	1295.8	1319.7	I	
55	<i>t</i>	287.1	337.1	387.1	437.1	487.1	537.1	587.1	<i>t</i>	
	V	7.78	8.40	9.00	9.59	10.16	10.73	11.30	V	55
	I	1175.4	1200.8	1225.6	1250.0	1274.2	1298.1	1322.0	I	
60	<i>t</i>	292.7	342.7	392.7	442.7	492.7	542.7	592.7	<i>t</i>	
	V	7.17	7.75	8.30	8.84	9.36	9.89	10.41	V	60
	I	1177.0	1202.6	1227.6	1252.1	1276.4	1300.4	1324.3	I	
65	<i>t</i>	298.0	348.0	398.0	448.0	498.0	548.0	598.0	<i>t</i>	
	V	6.65	7.20	7.70	8.20	8.69	9.17	9.65	V	65
	I	1178.5	1204.4	1229.5	1254.0	1278.4	1302.4	1326.4	I	
70	<i>t</i>	302.9	352.9	402.9	452.9	502.9	552.9	602.9	<i>t</i>	
	V	6.20	6.71	7.18	7.65	8.11	8.56	9.01	V	70
	I	1179.8	1205.9	1231.2	1255.8	1280.2	1304.3	1328.3	I	
75	<i>t</i>	307.6	357.6	407.6	457.6	507.6	557.6	607.6	<i>t</i>	
	V	5.81	6.28	6.73	7.17	7.60	8.02	8.44	V	75
	I	1181.1	1207.5	1232.8	1257.5	1282.0	1306.1	1330.1	I	
80	<i>t</i>	312.0	362.0	412.0	462.0	512.0	562.0	612.0	<i>t</i>	
	V	5.47	5.92	6.34	6.75	7.17	7.56	7.95	V	80
	I	1182.3	1208.8	1234.3	1259.0	1283.6	1307.8	1331.9	I	
85	<i>t</i>	316.3	366.3	416.3	466.3	516.3	566.3	616.3	<i>t</i>	
	V	5.16	5.59	5.99	6.38	6.76	7.14	7.51	V	85
	I	1183.4	1210.2	1235.8	1260.6	1285.2	1309.4	1333.5	I	

t = Temperature, deg. Fahr.

V = Specific volume, in cubic feet, per pound.

I = Total heat from water at 32 degrees, B Th.U.

PROPERTIES OF SUPERHEATED STEAM—Continued.

Pressure, Pounds Absolute.		Satur- ated Steam.	Degrees of Superheat.						Pressure, Pounds Absolute.	
			50	100	150	200	250	300		
90	<i>t</i>	320.3	370.3	420.3	470.3	520.3	570.3	620.3	<i>t</i>	90
	V	4.89	5.29	5.67	6.04	6.40	6.76	7.11	V	
	I	1184.4	1211.4	1237.2	1262.0	1286.6	1310.8	1334.9	I	
95	<i>t</i>	324.1	374.1	424.1	474.1	524.1	574.1	624.1	<i>t</i>	95
	V	4.65	5.03	5.39	5.74	6.09	6.43	6.76	V	
	I	1185.4	1212.6	1238.4	1263.4	1288.1	1312.3	1336.4	I	
100	<i>t</i>	327.8	377.8	427.8	477.8	527.8	577.8	627.8	<i>t</i>	100
	V	4.43	4.79	5.14	5.47	5.80	6.12	6.44	V	
	I	1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8	I	
105	<i>t</i>	331.4	381.4	431.4	481.4	531.4	581.4	631.4	<i>t</i>	105
	V	4.23	4.58	4.91	5.23	5.54	5.85	6.15	V	
	I	1187.2	1214.9	1240.8	1265.9	1290.6	1314.9	1339.1	I	
110	<i>t</i>	334.8	384.8	434.8	484.8	534.8	584.8	634.8	<i>t</i>	110
	V	4.05	4.38	4.70	5.01	5.31	5.61	5.90	V	
	I	1188.0	1215.9	1242.0	1267.1	1291.9	1316.2	1340.4	I	
115	<i>t</i>	338.1	388.1	438.1	488.1	538.1	588.1	638.1	<i>t</i>	115
	V	3.88	4.20	4.51	4.81	5.09	5.38	5.66	V	
	I	1188.8	1216.9	1243.1	1268.2	1293.0	1317.3	1341.5	I	
120	<i>t</i>	341.3	391.3	441.3	491.3	541.3	591.3	641.3	<i>t</i>	120
	V	3.73	4.04	4.33	4.62	4.89	5.17	5.44	V	
	I	1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7	I	
125	<i>t</i>	344.4	394.4	444.4	494.4	544.4	594.4	644.4	<i>t</i>	125
	V	3.58	3.88	4.17	4.45	4.71	4.97	5.23	V	
	I	1190.3	1218.8	1245.1	1270.4	1295.2	1319.5	1343.8	I	
130	<i>t</i>	347.4	397.4	447.4	497.4	547.4	597.4	647.4	<i>t</i>	130
	V	3.45	3.74	4.02	4.28	4.54	4.80	5.05	V	
	I	1191.0	1219.7	1246.1	1271.4	1296.2	1320.6	1344.9	I	
135	<i>t</i>	350.3	400.3	450.3	500.3	550.3	600.3	650.3	<i>t</i>	135
	V	3.33	3.61	3.88	4.14	4.38	4.63	4.87	V	
	I	1191.6	1220.6	1247.0	1272.3	1297.2	1321.6	1345.9	I	
140	<i>t</i>	353.1	403.1	453.1	503.1	553.1	603.1	653.1	<i>t</i>	140
	V	3.22	3.49	3.75	4.00	4.24	4.48	4.71	V	
	I	1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9	I	
145	<i>t</i>	355.8	405.8	455.8	505.8	555.8	605.8	655.8	<i>t</i>	145
	V	3.12	3.38	3.63	3.87	4.10	4.33	4.56	V	
	I	1192.8	1222.2	1248.8	1274.2	1299.1	1323.6	1347.9	I	
150	<i>t</i>	358.5	408.5	458.5	508.5	558.5	608.5	658.5	<i>t</i>	150
	V	3.01	3.27	3.51	3.75	3.97	4.19	4.41	V	
	I	1193.4	1223.0	1249.6	1275.1	1300.0	1324.5	1348.8	I	
155	<i>t</i>	361.0	411.0	461.0	511.0	561.0	611.0	661.0	<i>t</i>	155
	V	2.92	3.17	3.41	3.63	3.85	4.06	4.28	V	
	I	1194.0	1223.6	1250.5	1276.0	1300.8	1325.3	1349.7	I	
160	<i>t</i>	363.6	413.6	463.6	513.6	563.6	613.6	663.6	<i>t</i>	160
	V	2.83	3.07	3.30	3.53	3.74	3.95	4.15	V	
	I	1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6	I	
165	<i>t</i>	366.0	416.0	466.0	516.0	566.0	616.0	666.0	<i>t</i>	165
	V	2.75	2.99	3.21	3.43	3.64	3.84	4.04	V	
	I	1195.0	1225.2	1252.0	1277.6	1302.5	1327.1	1351.5	I	
170	<i>t</i>	368.5	418.5	468.5	518.5	568.5	618.5	668.5	<i>t</i>	170
	V	2.68	2.91	3.12	3.34	3.54	3.73	3.92	V	
	I	1195.4	1225.9	1252.8	1278.4	1303.3	1327.9	1352.3	I	

t = Temperature, deg. Fahr.

V = Specific volume, in cubic feet, per pound.

I = Total heat from water at 32 degrees, B.Th.U.

PROPERTIES OF SUPERHEATED STEAM—Continued.

Pressure, Pounds Absolute.		Saturated Steam.	Degrees of Superheat.						Pressure, Pounds Absolute.	
			50	100	150	200	250	300		
175	<i>t</i>	370.8	420.8	470.8	520.8	570.8	620.8	670.8	<i>t</i>	175
	V	2.60	2.83	3.04	3.24	3.44	3.63	3.82	V	
	I	1195.9	1226.6	1253.6	1279.1	1304.1	1328.7	1353.2	I	
180	<i>t</i>	373.1	423.1	473.1	523.1	573.1	623.1	673.1	<i>t</i>	180
	V	2.53	2.75	2.96	3.16	3.35	3.54	3.72	V	
	I	1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9	I	
185	<i>t</i>	375.4	425.4	475.4	525.4	575.4	625.4	675.4	<i>t</i>	185
	V	2.47	2.68	2.89	3.08	3.27	3.45	3.63	V	
	I	1196.8	1227.9	1255.0	1280.6	1305.6	1330.2	1354.7	I	
190	<i>t</i>	377.6	427.6	477.6	527.6	577.6	627.6	677.6	<i>t</i>	190
	V	2.41	2.62	2.81	3.00	3.19	3.37	3.55	V	
	I	1197.3	1228.6	1255.7	1281.3	1306.3	1330.9	1355.5	I	
195	<i>t</i>	379.8	429.8	479.8	529.8	579.8	629.8	679.8	<i>t</i>	195
	V	2.35	2.55	2.75	2.93	3.11	3.29	3.46	V	
	I	1197.7	1229.2	1256.4	1282.0	1307.0	1331.6	1356.2	I	
200	<i>t</i>	381.9	431.9	481.9	531.9	581.9	631.9	681.9	<i>t</i>	200
	V	2.29	2.49	2.68	2.86	3.04	3.21	3.38	V	
	I	1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0	I	
205	<i>t</i>	384.0	434.0	484.0	534.0	584.0	634.0	684.0	<i>t</i>	205
	V	2.24	2.44	2.62	2.80	2.97	3.14	3.30	V	
	I	1198.5	1230.4	1257.7	1283.3	1308.3	1333.0	1357.7	I	
210	<i>t</i>	386.0	436.0	486.0	536.0	586.0	636.0	686.0	<i>t</i>	210
	V	2.19	2.38	2.56	2.74	2.91	3.07	3.23	V	
	I	1198.8	1231.0	1258.4	1284.0	1309.0	1333.7	1358.4	I	
215	<i>t</i>	388.0	438.0	488.0	538.0	588.0	638.0	688.0	<i>t</i>	215
	V	2.14	2.33	2.51	2.68	2.84	3.00	3.16	V	
	I	1199.2	1231.6	1259.0	1284.6	1309.7	1334.4	1359.1	I	
220	<i>t</i>	389.9	439.9	489.9	539.9	589.9	639.9	689.9	<i>t</i>	220
	V	2.09	2.28	2.45	2.62	2.78	2.94	3.10	V	
	I	1199.6	1232.2	1259.6	1285.2	1310.3	1335.1	1359.8	I	
225	<i>t</i>	391.9	441.9	491.9	541.9	591.9	641.9	691.9	<i>t</i>	225
	V	2.05	2.23	2.40	2.57	2.72	2.88	3.03	V	
	I	1199.9	1232.7	1260.2	1285.9	1310.9	1335.7	1360.3	I	
230	<i>t</i>	393.8	443.8	493.8	543.8	593.8	643.8	693.8	<i>t</i>	230
	V	2.00	2.18	2.35	2.51	2.67	2.82	2.97	V	
	I	1200.2	1233.2	1260.7	1286.5	1311.0	1336.3	1361.0	I	
235	<i>t</i>	395.6	445.6	495.6	545.6	595.6	645.6	695.6	<i>t</i>	235
	V	1.96	2.14	2.30	2.46	2.62	2.77	2.91	V	
	I	1200.6	1233.8	1261.4	1287.1	1312.2	1337.0	1361.7	I	
240	<i>t</i>	397.4	447.4	497.4	547.4	597.4	647.4	697.4	<i>t</i>	240
	V	1.92	2.09	2.26	2.42	2.57	2.71	2.85	V	
	I	1200.9	1234.3	1261.9	1287.6	1312.8	1337.6	1362.3	I	
245	<i>t</i>	399.3	449.3	499.3	549.3	599.3	649.3	699.3	<i>t</i>	245
	V	1.89	2.05	2.22	2.37	2.52	2.66	2.80	V	
	I	1201.2	1234.8	1262.5	1288.2	1313.3	1338.2	1362.9	I	
250	<i>t</i>	401.0	451.0	501.0	551.0	601.0	651.0	701.0	<i>t</i>	250
	V	1.85	2.02	2.17	2.33	2.47	2.61	2.75	V	
	I	1201.5	1235.4	1263.0	1288.8	1313.9	1338.8	1363.5	I	
255	<i>t</i>	402.8	452.8	502.8	552.8	602.8	652.8	702.8	<i>t</i>	255
	V	1.81	1.98	2.14	2.28	2.43	2.56	2.70	V	
	I	1201.8	1235.9	1263.6	1289.3	1314.5	1339.3	1364.1	I	

t = Temperature, deg. Fahr.

V = Specific volume, in cubic feet, per pound.

I = Total heat from water at 32 degrees, B.Th.U.

THERMODYNAMICS APPLIED TO ENGINEERING

MEAN SPECIFIC HEAT OF SUPERHEATED STEAM

(Computed from Marks and Davis' Steam Tables).

Degrees of Superheat, Fahr.

	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
1	.452	.452	.453	.453	.454	.454	.454	.455	.455	.455	.455	.455	.455	.456	.456	.456	.456	.456	.456	.457
5	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460	.460
10	.465	.465	.465	.465	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.465	.465	.465	.465
15	.470	.470	.470	.470	.470	.470	.469	.469	.469	.469	.469	.469	.469	.469	.469	.468	.468	.468	.468	.468
20	.475	.475	.475	.475	.474	.474	.474	.474	.473	.473	.473	.473	.473	.472	.472	.472	.472	.472	.471	.471
25	.480	.480	.480	.479	.479	.479	.478	.478	.478	.477	.477	.477	.477	.477	.476	.476	.476	.475	.475	.475
30	.485	.485	.484	.484	.484	.483	.483	.483	.483	.482	.482	.481	.481	.481	.480	.480	.479	.479	.478	.477
35	.490	.490	.489	.489	.488	.488	.487	.487	.486	.486	.486	.485	.485	.484	.484	.483	.482	.482	.481	.480
40	.495	.494	.494	.493	.492	.492	.491	.491	.490	.490	.489	.489	.488	.488	.487	.486	.485	.484	.483	.482
50	.509	.508	.507	.506	.504	.503	.501	.500	.499	.498	.497	.496	.496	.495	.494	.493	.491	.490	.489	.487
75	.540	.535	.533	.530	.528	.525	.523	.521	.519	.517	.515	.513	.512	.511	.509	.506	.504	.502	.500	.497
100	.570	.560	.556	.552	.550	.546	.542	.540	.537	.534	.531	.528	.526	.524	.522	.518	.515	.512	.509	.505
125	.590	.585	.580	.575	.570	.565	.560	.556	.552	.548	.545	.542	.539	.537	.534	.532	.524	.520	.517	.510
150	.620	.615	.606	.597	.592	.585	.579	.572	.566	.562	.558	.554	.550	.547	.544	.539	.533	.528	.524	.518
175	.660	.645	.633	.622	.614	.605	.597	.589	.582	.577	.572	.566	.562	.558	.555	.548	.541	.535	.531	.524
200	.690	.675	.657	.645	.634	.623	.614	.605	.596	.590	.584	.578	.572	.568	.563	.556	.548	.542	.537	.529
225	.730	.710	.686	.672	.656	.643	.631	.620	.611	.603	.595	.589	.583	.578	.573	.562	.553	.547	.543	.535
250	.770	.740	.712	.695	.678	.663	.648	.635	.624	.615	.606	.599	.593	.587	.582	.570	.562	.555	.549	.540
275	.800	.770	.746	.722	.700	.681	.664	.650	.638	.629	.620	.610	.602	.596	.590	.578	.569	.561	.555	.545
300	.850	.805	.773	.740	.724	.702	.683	.666	.652	.641	.630	.621	.612	.605	.599	.586	.576	.568	.561	.550
350	.950	.890	.843	.802	.770	.741	.717	.697	.680	.666	.654	.642	.632	.624	.617	.602	.590	.580	.573	.560
400	1.06	.975	.913	.860	.818	.783	.754	.730	.709	.692	.677	.665	.654	.644	.635	.618	.604	.594	.585	.571
450	1.20	1.10	1.00	.925	.860	.816	.771	.742	.714	.690	.670	.653	.637	.624	.613	.602	.590	.580	.570	.556
500	1.30	1.15	1.07	.975	.920	.866	.830	.800	.766	.730	.707	.683	.662	.642	.624	.604	.590	.578	.568	.553
550	1.50	1.30	1.17	1.07	1.00	.933	.885	.837	.811	.780	.754	.741	.722	.707	.700	.670	.655	.636	.624	.607

Absolute Pressure, Pounds per Square Inch

TABLE OF THE THERMODYNAMIC
PROPERTIES OF THE SATURATED
VAPOUR OF AMMONIA

APPLIED TO ENGINEERING

SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE.*

Pressure (abs.), lbs./in. ²	Temp. °F.	Volume vapour, ft. ³ /lb.	Density vapour, lbs./ft. ³	Total Heat content.		Latent heat, B.Th.U. per lb.	Entropy.			Pressure (abs.), lbs./in. ²
				Liquid, B.Th.U. per lb.	Vapour, B.Th.U. per lb.		Liquid, B.Th.U. per lb. °F.	Evap., B.Th.U. per lb. °F.	Vapour, B.Th.U. per lb. °F.	
p	t	V_a	$1/V_a$	I_{liq}	I_a	L	ϕ_{liq}	L/T	ϕ_a	p
5.0	-63.11	49.31	0.02029	-24.5	588.3	612.8	-0.0599	1.5456	1.4857	5.0
5.5	-60.27	45.11	0.02217	-21.5	589.5	611.0	-0.0524	1.5301	1.4777	5.5
6.0	-57.64	41.59	0.02405	-18.7	590.6	609.3	-0.0455	1.5158	1.4703	6.0
6.5	-55.18	38.59	0.02591	-16.1	591.6	607.7	-0.0390	1.5026	1.4636	6.5
7.0	-52.88	36.01	0.02777	-13.7	592.5	606.2	-0.0330	1.4904	1.4574	7.0
7.5	-50.70	33.77	0.02962	-11.3	593.4	604.7	-0.0274	1.4790	1.4516	7.5
8.0	-48.64	31.79	0.03146	-9.2	594.2	603.4	-0.0221	1.4683	1.4462	8.0
8.5	-46.69	30.04	0.03329	-7.1	595.0	602.1	-0.0171	1.4582	1.4411	8.5
9.0	-44.83	28.48	0.03511	-5.1	595.7	600.8	-0.0123	1.4486	1.4363	9.0
9.5	-43.05	27.08	0.03693	-3.2	596.4	599.6	-0.0077	1.4396	1.4319	9.5
10.0	-41.34	25.81	0.03874	-1.4	597.1	598.5	-0.0034	1.4310	1.4276	10.0
10.5	-39.71	24.66	0.04055	+0.3	597.7	597.4	+0.0007	1.4228	1.4235	10.5
11.0	-38.14	23.61	0.04235	2.0	598.3	596.3	0.0047	1.4149	1.4196	11.0
11.5	-36.62	22.65	0.04414	3.6	598.9	595.3	0.0085	1.4074	1.4159	11.5
12.0	-35.16	21.77	0.04593	5.1	599.4	594.3	0.0122	1.4002	1.4124	12.0
12.5	-33.74	20.96	0.04772	6.7	600.0	593.3	0.0157	1.3933	1.4090	12.5
13.0	-32.37	20.20	0.04950	8.1	600.5	592.4	0.0191	1.3866	1.4057	13.0
13.5	-31.05	19.50	0.05128	9.6	601.0	591.4	0.0225	1.3801	1.4026	13.5
14.0	-29.76	18.85	0.05305	10.9	601.4	590.5	0.0257	1.3739	1.3996	14.0
14.5	-28.51	18.24	0.05482	12.2	601.9	589.7	0.0288	1.3679	1.3967	14.5
15.0	-27.29	17.67	0.05658	13.6	602.4	588.8	0.0318	1.3620	1.3938	15.0
15.5	-26.11	17.14	0.05834	14.8	602.8	588.0	0.0347	1.3564	1.3911	15.5
16.0	-24.95	16.64	0.06010	16.0	603.2	587.2	0.0375	1.3510	1.3885	16.0
16.5	-23.83	16.17	0.06186	17.2	603.6	586.4	0.0403	1.3456	1.3859	16.5
17.0	-22.73	15.72	0.06361	18.4	604.0	585.6	0.0430	1.3405	1.3835	17.0
17.5	-21.66	15.30	0.06535	19.6	604.4	584.8	0.0456	1.3354	1.3810	17.5
18.0	-20.61	14.90	0.06710	20.7	604.8	584.1	0.0482	1.3305	1.3787	18.0
18.5	-19.59	14.53	0.06884	21.8	605.1	583.3	0.0507	1.3258	1.3765	18.5
19.0	-18.58	14.17	0.07058	22.9	605.5	582.6	0.0531	1.3211	1.3742	19.0
19.5	-17.60	13.83	0.07232	23.9	605.8	581.9	0.0555	1.3166	1.3721	19.5
20.0	-16.64	13.50	0.07405	25.0	606.2	581.2	0.0578	1.3122	1.3700	20.0
20.5	-15.70	13.20	0.07578	26.0	606.5	580.5	0.0601	1.3078	1.3679	20.5
21.0	-14.78	12.90	0.07751	27.0	606.8	579.8	0.0623	1.3036	1.3659	21.0
21.5	-13.87	12.62	0.07924	27.9	607.1	579.2	0.0645	1.2995	1.3640	21.5
22.0	-12.98	12.35	0.08096	28.9	607.4	578.5	0.0666	1.2955	1.3621	22.0
22.5	-12.11	12.09	0.08268	29.8	607.7	577.9	0.0687	1.2915	1.3602	22.5
23.0	-11.25	11.85	0.08440	30.8	608.1	577.3	0.0708	1.2876	1.3584	23.0
23.5	-10.41	11.61	0.08612	31.7	608.3	576.6	0.0728	1.2838	1.3566	23.5
24.0	-9.58	11.39	0.08783	32.6	608.6	576.0	0.0748	1.2801	1.3549	24.0
24.5	-8.76	11.17	0.08955	33.5	608.9	575.4	0.0768	1.2764	1.3532	24.5
25.0	-7.96	10.96	0.09126	34.3	609.1	574.8	0.0787	1.2728	1.3515	25.0
25.5	-7.17	10.76	0.09297	35.2	609.4	574.2	0.0805	1.2693	1.3498	25.5
26.0	-6.39	10.56	0.09468	36.0	609.7	573.7	0.0824	1.2658	1.3482	26.0
26.5	-5.63	10.38	0.09638	36.8	609.9	573.1	0.0842	1.2625	1.3467	26.5
27.0	-4.87	10.20	0.09809	37.7	610.2	572.5	0.0860	1.2591	1.3451	27.0
27.5	-4.13	10.02	0.09979	38.4	610.4	572.0	0.0878	1.2558	1.3436	27.5
28.0	-3.40	9.853	0.1015	39.3	610.7	571.4	0.0895	1.2526	1.3421	28.0
28.5	-2.68	9.691	0.1032	40.0	610.9	570.9	0.0912	1.2494	1.3406	28.5
29.0	-1.97	9.534	0.1049	40.8	611.1	570.3	0.0929	1.2463	1.3392	29.0
29.5	-1.27	9.383	0.1066	41.6	611.4	569.8	0.0945	1.2433	1.3378	29.5
30.0	-0.57	9.236	0.1083	42.3	611.6	569.3	0.0962	1.2402	1.3364	30.0

* By courtesy of the U.S. Bureau of Standards.

SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE—Continued.

Pressure (abs.), lbs./in. ²	Temp. °F.	Volume vapour, ft ³ /lb.	Density vapour, lbs./ft. ³	Total Heat content.		Latent heat, B.Th.U. per lb.	Entropy.			Pressure (abs.), lbs./in. ²
				Liquid, B.Th.U. per lb.	Vapour, B.Th.U. per lb.		Liquid, B.Th.U. per lb. °F.	Evap. B.Th.U. per lb. °F.	Vapour, B.Th.U. per lb. °F.	
<i>p</i>	<i>t</i>	<i>V_a</i>	<i>1/V_a</i>	<i>l_{liq}</i>	<i>l_v</i>	<i>L</i>	<i>φ_{liq}</i>	<i>L/T</i>	<i>φ_v</i>	<i>p</i>
30	— 0·57	9·236	0·1083	42·3	611·6	569·3	0·0962	1·2402	1·3364	30
31	+ 0·79	8·955	0·1117	43·8	612·0	568·2	0·0993	1·2343	1·3336	31
32	2·11	8·693	0·1150	45·2	612·4	567·2	0·1024	1·2286	1·3310	32
33	3·40	8·445	0·1184	46·6	612·8	566·2	0·1055	1·2230	1·3285	33
34	4·66	8·211	0·1218	48·0	613·2	565·2	0·1084	1·2176	1·3260	34
35	5·89	7·991	0·1251	49·3	613·6	564·3	0·1113	1·2123	1·3236	35
36	7·09	7·782	0·1285	50·6	614·0	563·4	0·1141	1·2072	1·3213	36
37	8·27	7·584	0·1319	51·9	614·3	562·4	0·1168	1·2022	1·3190	37
38	9·42	7·396	0·1352	53·2	614·7	561·5	0·1195	1·1973	1·3168	38
39	10·55	7·217	0·1386	54·4	615·0	560·6	0·1221	1·1925	1·3146	39
40	11·66	7·047	0·1419	55·6	615·4	559·8	0·1246	1·1879	1·3125	40
41	12·74	6·885	0·1452	56·8	615·7	558·9	0·1271	1·1833	1·3104	41
42	13·81	6·731	0·1486	57·9	616·0	558·1	0·1296	1·1788	1·3084	42
43	14·85	6·583	0·1519	59·1	616·3	557·2	0·1320	1·1745	1·3065	43
44	15·88	6·442	0·1552	60·2	616·6	556·4	0·1343	1·1703	1·3046	44
45	16·88	6·307	0·1586	61·3	616·9	555·6	0·1366	1·1661	1·3027	45
46	17·87	6·177	0·1619	62·4	617·2	554·8	0·1389	1·1620	1·3009	46
47	18·84	6·053	0·1652	63·4	617·4	554·0	0·1411	1·1580	1·2991	47
48	19·80	5·934	0·1685	64·5	617·7	553·2	0·1433	1·1540	1·2973	48
49	20·74	5·820	0·1718	65·5	618·0	552·5	0·1454	1·1502	1·2956	49
50	21·67	5·710	0·1751	66·5	618·2	551·7	0·1475	1·1464	1·2939	50
51	22·58	5·604	0·1785	67·5	618·5	551·0	0·1496	1·1427	1·2923	51
52	23·48	5·502	0·1818	68·5	618·7	550·2	0·1516	1·1390	1·2906	52
53	24·36	5·404	0·1851	69·5	619·0	549·5	0·1536	1·1354	1·2890	53
54	25·23	5·309	0·1884	70·4	619·2	548·8	0·1556	1·1319	1·2875	54
55	26·09	5·218	0·1917	71·4	619·4	548·0	0·1575	1·1284	1·2859	55
56	26·94	5·129	0·1950	72·3	619·7	547·4	0·1594	1·1250	1·2844	56
57	27·77	5·044	0·1983	73·3	619·9	546·6	0·1613	1·1217	1·2830	57
58	28·59	4·962	0·2015	74·2	620·1	545·9	0·1631	1·1184	1·2815	58
59	29·41	4·882	0·2048	75·0	620·3	545·3	0·1650	1·1151	1·2801	59
60	30·21	4·805	0·2081	75·9	620·5	544·6	0·1668	1·1119	1·2787	60
61	31·00	4·730	0·2114	76·8	620·7	543·9	0·1685	1·1088	1·2773	61
62	31·78	4·658	0·2147	77·7	620·9	543·2	0·1703	1·1056	1·2759	62
63	32·55	4·588	0·2180	78·5	621·1	542·6	0·1720	1·1026	1·2746	63
64	33·31	4·519	0·2213	79·4	621·3	541·9	0·1737	1·0996	1·2733	64
65	34·06	4·453	0·2245	80·2	621·5	541·3	0·1754	1·0966	1·2720	65
66	34·81	4·389	0·2278	81·0	621·7	540·7	0·1770	1·0937	1·2707	66
67	35·54	4·327	0·2311	81·8	621·9	540·1	0·1787	1·0907	1·2694	67
68	36·27	4·267	0·2344	82·6	622·0	539·4	0·1803	1·0879	1·2682	68
69	36·99	4·208	0·2377	83·4	622·2	538·8	0·1819	1·0851	1·2670	69
70	37·70	4·151	0·2409	84·2	622·4	538·2	0·1835	1·0823	1·2658	70
71	38·40	4·095	0·2442	85·0	622·6	537·6	0·1850	1·0795	1·2645	71
72	39·09	4·041	0·2475	85·8	622·8	537·0	0·1866	1·0768	1·2634	72
73	39·78	3·988	0·2507	86·5	622·9	536·4	0·1881	1·0741	1·2622	73
74	40·46	3·937	0·2540	87·3	623·1	535·8	0·1896	1·0715	1·2611	74
75	41·13	3·887	0·2573	88·0	623·2	535·2	0·1910	1·0689	1·2599	75
76	41·80	3·838	0·2606	88·8	623·4	534·6	0·1925	1·0663	1·2588	76
77	42·46	3·790	0·2638	89·5	623·5	534·0	0·1940	1·0637	1·2577	77
78	43·11	3·744	0·2671	90·2	623·7	533·5	0·1954	1·0612	1·2566	78
79	43·76	3·699	0·2704	90·9	623·8	532·9	0·1968	1·0587	1·2555	79
80	44·40	3·655	0·2736	91·7	624·0	532·3	0·1982	1·0563	1·2545	80

SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE—Continued.

Pressure (abs.). lbs./in. ²	Temp. °F.	Volume vapour. ft. ³ /lb.	Density vapour. lbs./ft. ³	Total Heat content.		Latent heat. B.Th.U. per lb.	Entropy.			Pressure (abs.). lbs./in. ²
				Liquid. B.Th.U. per lb.	Vapour. B.Th.U. per lb.		Liquid. B.Th.U. per lb. °F.	Evap. B.Th.U. per lb. °F.	Vapour. B.Th.U. per lb. °F.	
<i>p</i>	<i>t</i>	<i>V_a</i>	<i>1/V_a</i>	<i>I_{liq}</i>	<i>I_a</i>	<i>L</i>	<i>φ_{liq}</i>	<i>L/T</i>	<i>φ_v</i>	<i>p</i>
80	44.40	3.655	0.2736	91.7	624.0	532.3	0.1982	1.0563	1.2545	80
81	45.03	3.612	0.2769	92.4	624.1	531.7	0.1996	1.0538	1.2534	81
82	45.66	3.570	0.2801	93.1	624.3	531.2	0.2010	1.0514	1.2524	82
83	46.28	3.528	0.2834	93.8	624.4	530.6	0.2024	1.0490	1.2514	83
84	46.89	3.488	0.2867	94.5	624.6	530.1	0.2037	1.0467	1.2504	84
85	47.50	3.449	0.2899	95.1	624.7	529.6	0.2051	1.0443	1.2494	85
86	48.11	3.411	0.2932	95.8	624.8	529.0	0.2064	1.0420	1.2484	86
87	48.71	3.373	0.2964	96.5	625.0	528.5	0.2077	1.0397	1.2474	87
88	49.30	3.337	0.2997	97.2	625.1	527.9	0.2090	1.0375	1.2465	88
89	49.89	3.301	0.3030	97.8	625.2	527.4	0.2103	1.0352	1.2455	89
90	50.47	3.266	0.3062	98.4	625.3	526.9	0.2115	1.0330	1.2445	90
91	51.05	3.231	0.3095	99.1	625.5	526.4	0.2128	1.0308	1.2436	91
92	51.62	3.198	0.3127	99.8	625.6	525.8	0.2141	1.0286	1.2427	92
93	52.19	3.165	0.3160	100.4	625.7	525.3	0.2153	1.0265	1.2418	93
94	52.76	3.132	0.3192	101.0	625.8	524.8	0.2165	1.0243	1.2408	94
95	53.32	3.101	0.3225	101.6	625.9	524.3	0.2177	1.0222	1.2399	95
96	53.87	3.070	0.3258	102.3	626.1	523.8	0.2190	1.0201	1.2391	96
97	54.42	3.039	0.3290	102.9	626.2	523.3	0.2201	1.0181	1.2382	97
98	54.97	3.010	0.3323	103.5	626.3	522.8	0.2213	1.0160	1.2373	98
99	55.51	2.980	0.3355	104.1	626.4	522.3	0.2225	1.0140	1.2365	99
100	56.05	2.952	0.3388	104.7	626.5	521.8	0.2237	1.0119	1.2356	100
102	57.11	2.896	0.3453	105.9	626.7	520.8	0.2260	1.0079	1.2339	102
104	58.16	2.843	0.3518	107.1	626.9	519.8	0.2282	1.0041	1.2323	104
106	59.19	2.791	0.3583	108.3	627.1	518.8	0.2305	1.0002	1.2307	106
108	60.21	2.741	0.3648	109.4	627.3	517.9	0.2327	0.9964	1.2291	108
110	61.21	2.693	0.3713	110.5	627.5	517.0	0.2348	0.9927	1.2275	110
112	62.20	2.647	0.3778	111.7	627.7	516.0	0.2369	0.9890	1.2259	112
114	63.17	2.602	0.3843	112.8	627.9	515.1	0.2390	0.9854	1.2244	114
116	64.13	2.559	0.3909	113.9	628.1	514.2	0.2411	0.9819	1.2230	116
118	65.08	2.517	0.3974	114.9	628.2	513.3	0.2431	0.9784	1.2215	118
120	66.02	2.476	0.4039	116.0	628.4	512.4	0.2452	0.9749	1.2201	120
122	66.94	2.437	0.4104	117.1	628.6	511.5	0.2471	0.9715	1.2186	122
124	67.86	2.399	0.4169	118.1	628.7	510.6	0.2491	0.9682	1.2173	124
126	68.76	2.362	0.4234	119.1	628.9	509.8	0.2510	0.9649	1.2159	126
128	69.65	2.326	0.4299	120.1	629.0	508.9	0.2529	0.9616	1.2145	128
130	70.53	2.291	0.4364	121.1	629.2	508.1	0.2548	0.9584	1.2132	130
132	71.40	2.258	0.4429	122.1	629.3	507.2	0.2567	0.9552	1.2119	132
134	72.26	2.225	0.4494	123.1	629.5	506.4	0.2585	0.9521	1.2106	134
136	73.11	2.193	0.4559	124.1	629.6	505.5	0.2603	0.9490	1.2093	136
138	73.95	2.162	0.4624	125.1	629.8	504.7	0.2621	0.9460	1.2081	138
140	73.79	2.132	0.4690	126.0	629.9	503.9	0.2638	0.9430	1.2068	140
142	75.61	2.103	0.4755	126.9	630.0	503.1	0.2656	0.9400	1.2056	142
144	76.42	2.075	0.4820	127.9	630.2	502.3	0.2673	0.9371	1.2044	144
146	77.23	2.047	0.4885	128.8	630.3	501.5	0.2690	0.9342	1.2032	146
148	78.03	2.020	0.4951	129.7	630.4	500.7	0.2707	0.9313	1.2020	148
150	78.81	1.994	0.5016	130.6	630.5	499.9	0.2724	0.9285	1.2009	150

THERMODYNAMICS APPLIED TO ENGINEERING

SATURATED AMMONIA: ABSOLUTE PRESSURE TABLE—*Continued.*

Pressure (abs.), lbs./in. ²	Temp. °F.	Volume vapour, ft. ³ /lb.	Density vapour, lbs./ft. ³	Total Heat content.		Latent heat, B.Th.U. per lb.	Entropy.			Pressure (abs.), lbs./in. ²
				Liquid, B.Th.U. per lb.	Vapour, B.Th.U. per lb.		Liquid, B.Th.U. per lb., °F.	Evap. B.Th.U. per lb., °F.	Vapour, B.Th.U. per lb., °F.	
<i>p</i>	<i>t</i>	<i>V_g</i>	<i>1/V_g</i>	<i>I_{liq}</i>	<i>I_v</i>	<i>L</i>	<i>φ_{liq}</i>	<i>L/T</i>	<i>φ_v</i>	<i>p</i>
150	78·81	1·994	0·5016	130·6	630·5	499·9	0·2724	0·9285	1·2009	150
152	79·60	1·968	0·5081	131·5	630·6	499·1	0·2740	0·9257	1·1997	152
154	80·37	1·943	0·5147	132·4	630·7	498·3	0·2756	0·9229	1·1985	154
156	81·13	1·919	0·5212	133·3	630·9	497·6	0·2772	0·9202	1·1974	156
158	81·89	1·895	0·5277	134·2	631·0	496·8	0·2788	0·9175	1·1963	158
160	82·64	1·872	0·5343	135·0	631·1	496·1	0·2804	0·9148	1·1952	160
162	83·39	1·849	0·5408	135·9	631·2	495·3	0·2820	0·9122	1·1942	162
164	84·12	1·827	0·5473	136·8	631·3	494·5	0·2835	0·9096	1·1931	164
166	84·85	1·805	0·5539	137·6	631·4	493·8	0·2850	0·9070	1·1920	166
168	85·57	1·784	0·5604	138·4	631·5	493·1	0·2866	0·9044	1·1910	168
170	86·29	1·764	0·5670	139·3	631·6	492·3	0·2881	0·9019	1·1900	170
172	87·00	1·744	0·5735	140·1	631·7	491·6	0·2895	0·8994	1·1889	172
174	87·71	1·724	0·5801	140·9	631·7	490·8	0·2910	0·8969	1·1879	174
176	88·40	1·705	0·5866	141·7	631·8	490·1	0·2925	0·8944	1·1869	176
178	89·10	1·686	0·5932	142·5	631·9	489·4	0·2939	0·8920	1·1859	178
180	89·78	1·667	0·5998	143·3	632·0	488·7	0·2954	0·8896	1·1850	180
182	90·46	1·649	0·6063	144·1	632·1	488·0	0·2968	0·8872	1·1840	182
184	91·14	1·632	0·6129	144·8	632·1	487·3	0·2982	0·8848	1·1830	184
186	91·80	1·614	0·6195	145·6	632·2	486·6	0·2996	0·8825	1·1821	186
188	92·47	1·597	0·6261	146·4	632·3	485·9	0·3010	0·8801	1·1811	188
190	93·13	1·581	0·6326	147·2	632·4	485·2	0·3024	0·8778	1·1802	190
192	93·78	1·564	0·6392	147·9	632·4	484·5	0·3037	0·8755	1·1792	192
194	94·43	1·548	0·6458	148·7	632·5	483·8	0·3050	0·8733	1·1783	194
196	95·07	1·533	0·6524	149·5	632·6	483·1	0·3064	0·8710	1·1774	196
198	95·71	1·517	0·6590	150·2	632·6	482·4	0·3077	0·8688	1·1765	198
200	96·34	1·502	0·6656	150·9	632·7	481·8	0·3090	0·8666	1·1756	200
205	97·00	1·466	0·6821	152·7	632·8	480·1	0·3122	0·8612	1·1734	205
210	99·43	1·431	0·6986	154·6	633·0	478·4	0·3154	0·8559	1·1713	210
215	100·94	1·398	0·7152	156·3	633·1	476·8	0·3185	0·8507	1·1692	215
220	102·42	1·367	0·7318	158·0	633·2	475·2	0·3216	0·8455	1·1671	220
225	103·87	1·336	0·7484	159·7	633·3	473·6	0·3246	0·8405	1·1651	225
230	105·30	1·307	0·7650	161·4	633·4	472·0	0·3275	0·8356	1·1631	230
235	106·71	1·279	0·7817	163·1	633·5	470·4	0·3304	0·8307	1·1611	235
240	108·09	1·253	0·7984	164·7	633·6	468·9	0·3332	0·8260	1·1592	240
245	109·46	1·227	0·8151	166·4	633·8	467·3	0·3360	0·8213	1·1573	245
250	110·80	1·202	0·8319	168·0	633·8	465·8	0·3388	0·8167	1·1555	250
255	112·12	1·178	0·8487	169·5	633·8	464·3	0·3415	0·8121	1·1536	255
260	113·42	1·155	0·8655	171·1	633·9	462·8	0·3441	0·8077	1·1518	260
265	114·71	1·133	0·8824	172·6	633·9	461·3	0·3468	0·8033	1·1501	265
270	115·97	1·112	0·8993	174·1	633·9	459·8	0·3494	0·7989	1·1483	270
275	117·22	1·091	0·9162	175·6	634·0	458·4	0·3519	0·7947	1·1466	275
280	118·45	1·072	0·9332	177·1	634·0	456·9	0·3545	0·7904	1·1449	280
285	119·66	1·052	0·9502	178·6	634·0	455·4	0·3569	0·7863	1·1432	285
290	120·86	1·034	0·9672	180·0	634·0	454·0	0·3594	0·7821	1·1415	290
295	122·05	1·016	0·9843	181·5	634·0	452·5	0·3618	0·7781	1·1399	295
300	123·21	0·999	1·0015	182·9	634·0	451·1	0·3642	0·7741	1·1383	300

TABLE OF THE THERMODYNAMIC
PROPERTIES OF SATURATED
MERCURY VAPOUR

THERMODYNAMICS APPLIED TO ENGINEERING

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PROPERTIES OF SATURATED MERCURY VAPOUR. (L. A. SHELDON.)*
(Abridged and altered.)

	Pressure, lbs./in. ²	Tempera- ture, deg. Fahr.	Heat of Liq. B.Th.U.	Heat of Vap. B.Th.U.	Total Heat, B.Th.U.	Entropy of Liq.	Entropy of Vap.	Specific Vol. cu. ft. per pound.	Density, pounds per cu. ft.
	<i>p</i>	<i>t</i>	<i>l</i> _{liq}	<i>L</i>	<i>l</i> _m	<i>φ</i> _{liq}	<i>φ</i> _{vap}	<i>V</i> _m	<i>W</i> _m
(29 in.)	0.4	402.3	12.3	128.2	140.5	0.0185	0.1487	114.5	0.008733
	0.49	414.0	12.5	127.9	140.5	0.0190	0.1464	94.63	0.01057
	0.5	415.2	12.6	127.9	140.6	0.0192	0.1462	92.85	0.01077
	0.6	426.1	13.0	127.6	140.6	0.0195	0.1441	78.23	0.01278
(28.5 in.)	0.7	435.4	13.3	127.4	140.7	0.0198	0.1424	67.70	0.01477
	0.735	438.4	13.4	127.4	140.8	0.0199	0.1418	64.64	0.01547
	0.8	443.8	13.5	127.2	140.8	0.0201	0.1409	59.72	0.01675
(28 in.)	0.9	451.4	13.8	127.1	140.9	0.0203	0.1395	53.47	0.01870
	0.98	456.8	14.1	126.9	141.0	0.0205	0.1385	49.37	0.02025
	1.0	458.1	14.1	126.9	141.0	0.0206	0.1383	48.45	0.02064
	1.1	464.3	14.2	126.8	141.0	0.0208	0.1372	44.29	0.02258
	1.2	470.0	14.5	126.7	141.1	0.0210	0.1362	40.83	0.02449
	1.3	475.4	14.7	126.5	141.2	0.0212	0.1353	37.88	0.02640
	1.4	480.5	14.8	126.4	141.2	0.0214	0.1345	35.37	0.02827
	1.5	485.1	15.0	126.3	141.3	0.0215	0.1337	33.14	0.03017
	1.6	489.6	15.1	126.2	141.3	0.0217	0.1330	31.21	0.03204
	1.7	493.7	15.2	126.1	141.3	0.0219	0.1323	29.47	0.03393
	1.8	497.7	15.3	126.1	141.4	0.0220	0.1317	27.94	0.03579
	1.9	501.5	15.5	126.0	141.5	0.0221	0.1311	26.56	0.03765
	2.0	505.2	15.6	125.9	141.5	0.0223	0.1305	25.32	0.03949
	3.0	535.4	16.6	125.2	141.8	0.0233	0.1258	17.34	0.05767
	4.0	558.0	17.4	124.7	142.1	0.0240	0.1226	13.26	0.07540
	5.0	576.2	18.0	124.3	142.3	0.0246	0.1200	10.77	0.09281
	6.0	591.4	18.5	124.0	142.5	0.0251	0.1179	9.096	0.1099
	7.0	605.0	18.9	123.7	142.6	0.0256	0.1162	7.883	0.1269
	8.0	616.8	19.3	123.4	142.7	0.0259	0.1147	6.963	0.1436
	9.0	627.5	19.7	123.2	142.9	0.0262	0.1133	6.245	0.1601
	10.0	637.3	20.0	123.0	143.0	0.0265	0.1121	5.661	0.1766
	11.0	646.3	20.3	122.8	143.1	0.0267	0.1110	5.181	0.1930
	12.0	654.6	20.6	122.6	143.2	0.0270	0.1100	4.781	0.2092
	13.0	662.3	20.8	122.4	143.3	0.0272	0.1091	4.439	0.2253
	14.0	669.7	21.1	122.3	143.4	0.0274	0.1083	4.144	0.2413
	15.0	676.5	21.3	122.1	143.4	0.0276	0.1075	3.892	0.2569
	16.0	683.0	21.5	122.0	143.5	0.0278	0.1067	3.667	0.2727
	17.0	689.2	21.7	121.8	143.5	0.0280	0.1060	3.466	0.2885
	18.0	695.0	21.9	121.7	143.6	0.0282	0.1054	3.288	0.3042
	19.0	700.8	22.1	121.6	143.7	0.0283	0.1048	3.127	0.3198
	20.0	706.2	22.2	121.5	143.7	0.0285	0.1042	2.983	0.3352
	21.0	711.3	22.4	121.4	143.8	0.0286	0.1036	2.852	0.3506
	22.0	716.4	22.6	121.2	143.8	0.0287	0.1031	2.733	0.3659
	23.0	721.2	22.8	121.1	143.9	0.0289	0.1026	2.623	0.3813
	24.0	725.8	22.9	121.0	143.9	0.0290	0.1021	2.522	0.3966
	25.0	730.4	23.0	120.9	143.9	0.0292	0.1016	2.429	0.4117
	26.0	734.6	23.2	120.8	144.0	0.0293	0.1012	2.342	0.4270
	27.0	738.8	23.3	120.8	144.1	0.0294	0.1008	2.262	0.4420
	28.0	742.9	23.5	120.7	144.2	0.0295	0.1003	2.187	0.4572
	29.0	747.0	23.6	120.6	144.2	0.0296	0.09992	2.118	0.4722

* By courtesy of Am. Soc. of Mech. Eng.

THERMODYNAMICS APPLIED TO ENGINEERING

PROPERTIES OF SATURATED MERCURY VAPOUR—*Continued.*

Pressure, lbs. in. ²	Tempera- ture, deg. Fahr.	Heat of Liq. B.Th.U.	Heat of Vap. B.Th.U.	Total Heat, B.Th.U.	Entropy of Liq.	Entropy of Vap.	Specific Vol. cu. ft. per pound.	Density, pounds per cu. ft.
p	t	l_{liq}	L	l_m	ϕ'_{liq}	ϕ_{vap}	V_m	W_m
30.0	750.0	23.8	120.5	144.3	0.0297	0.09953	2.053	0.4871
31.0	754.7	23.9	120.4	144.3	0.0298	0.09915	1.992	0.5020
32.0	758.3	24.0	120.3	144.3	0.0299	0.09879	1.934	0.5170
33.0	761.8	24.1	120.2	144.3	0.0300	0.09844	1.881	0.5316
34.0	765.5	24.2	120.2	144.4	0.0301	0.09808	1.830	0.5465
35.0	769.0	24.3	120.1	144.4	0.0302	0.09774	1.782	0.5613
36.0	772.3	24.4	120.0	144.4	0.0303	0.09742	1.737	0.5758
37.0	775.5	24.5	119.9	144.4	0.0304	0.09710	1.694	0.5905
38.0	778.6	24.6	119.9	144.5	0.0305	0.09680	1.652	0.6052
39.0	781.8	24.7	119.8	144.5	0.0306	0.09650	1.613	0.6198
40.0	784.8	24.8	119.7	144.5	0.0307	0.09622	1.576	0.6344
41.0	787.8	24.9	119.7	144.6	0.0307	0.09593	1.541	0.6491
42.0	790.7	25.0	119.6	144.6	0.0308	0.09567	1.507	0.6635
43.0	793.6	25.1	119.5	144.6	0.0309	0.09539	1.475	0.6781
44.0	796.5	25.2	119.5	144.7	0.0310	0.09511	1.444	0.6925
45.0	799.3	25.3	119.4	144.7	0.0311	0.09487	1.415	0.7069
50.0	812.5	25.8	119.1	144.9	0.0314	0.09365	1.284	0.7788
55.0	824.6	26.2	118.9	145.1	0.0317	0.09255	1.177	0.8498
60.0	836.1	26.6	118.6	145.2	0.0320	0.09154	1.087	0.9204
65.0	847.0	26.9	118.4	145.3	0.0322	0.09060	1.010	0.9901
70.0	856.6	27.2	118.2	145.4	0.0325	0.08977	0.9440	1.060
75.0	866.0	27.5	118.0	145.5	0.0327	0.08897	0.8860	1.129
80.0	874.8	27.8	117.8	145.6	0.0330	0.08824	0.8349	1.198
85.0	883.7	28.1	117.6	145.7	0.0332	0.08751	0.7901	1.266
90.0	891.6	28.5	117.4	145.8	0.0334	0.08687	0.7497	1.334
100.0	906.9	28.9	117.1	145.9	0.0337	0.08565	0.6811	1.468
110.0	921.1	29.3	116.7	146.0	0.0341	0.08454	0.6242	1.602
120.0	934.4	29.8	116.4	146.2	0.0344	0.08353	0.5767	1.734
130.0	946.7	30.2	116.2	146.4	0.0347	0.08261	0.5360	1.866
140.0	958.3	30.6	115.9	146.5	0.0350	0.08175	0.5012	1.995
150.0	969.4	30.9	115.7	146.6	0.0353	0.08094	0.4706	2.125
160.0	979.9	31.3	115.4	146.7	0.0355	0.08019	0.4438	2.253
170.0	989.9	31.6	115.2	146.8	0.0356	0.07949	0.4200	2.381
180.0	999.6	31.9	115.0	146.9	0.0357	0.07881	0.3990	2.506

TABLE OF THE THERMODYNAMIC
PROPERTIES OF CARBON DIOXIDE

THERMODYNAMICS

APPLIED TO ENGINEERING

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PROPERTIES OF CARBON DIOXIDE.*

Temp. deg. Fahr.	Pres- sure pounds per sq. in.	Specific volume.		Heat content.		Heat of Vaporiza- tion.	Entropy.	
		Liquid	Vapour	Liquid.	Vapour.		Liquid.	Vapor- ization.
		cu. ft. per lb.	cu. ft. per lb.					
<i>t</i>	<i>p</i>	<i>V_{liq}</i>	<i>V_{vap}</i>	<i>I_{liq}</i>	<i>I_{cd}</i>	<i>L</i>	<i>φ_{liq}</i>	<i>φ_{vap}</i> = $\frac{L}{T}$
-25	203.4	0.01551	0.4575	-26.91	100.22	127.13	-0.0561	0.2925
-20	221.0	0.01556	0.4173	-24.75	100.50	125.25	-0.0513	0.2851
-15	240.5	0.01565	0.3810	-22.72	100.74	123.46	-0.0467	0.2778
-10	261.8	0.01578	0.3481	-20.56	100.88	121.44	-0.0419	0.2702
-5	284.1	0.01594	0.3185	-18.31	100.99	119.30	-0.0372	0.2626
0	308.0	0.01612	0.2918	-16.00	101.00	117.00	-0.0325	0.2549
5	334.2	0.01631	0.2672	-13.73	100.97	114.70	-0.0276	0.2470
10	362.5	0.01652	0.2450	-11.36	100.89	112.35	-0.0227	0.2391
15	391.0	0.01675	0.2244	-8.94	100.70	109.64	-0.0176	0.2309
20	421.6	0.01696	0.2060	-6.40	100.43	106.83	-0.0126	0.2228
25	454.7	0.01720	0.1882	-3.74	100.08	102.82	-0.0074	0.2143
30	488.8	0.01747	0.1724	-1.04	99.43	100.47	-0.0021	0.2044
35	525.5	0.01776	0.1580	+1.74	99.00	97.26	+0.0032	0.1968
40	564.5	0.01805	0.1444	4.36	98.25	93.89	0.0087	0.1876
45	606.0	0.01835	0.1323	7.54	97.32	89.78	0.0145	0.1780
50	650.0	0.01870	0.1205	10.76	96.30	85.54	0.0205	0.1679
55	696.0	0.01918	0.1090	14.18	95.00	80.82	0.0268	0.1572
60	744.0	0.01986	0.0986	17.85	93.54	75.69	0.0334	0.1461
65	794.0	0.02052	0.0890	21.50	91.67	70.17	0.0406	0.1334
70	847.0	0.02136	0.0816	26.02	89.35	63.33	0.0483	0.1201
75	906.0	0.02230	0.0766	30.96	86.36	55.40	0.0576	0.1037
80	965.0	0.02365	0.0614	36.80	82.80	46.00	0.0684	0.0843
85	1026.0	0.02620	0.0500	44.67	76.60	30.23	0.0828	0.0559
87	1052.0	0.02782	0.0440	48.98	71.80	23.82	0.0923	0.0393
88.43	1071.0	0.0346	0.0346	61.45	61.45	0.00	0.1120	0.000

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MATHEMATICAL TABLES

THERMODYNAMICS APPLIED TO ENGINEERING TRIGONOMETRIC FUNCTIONS.*

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Angle.		Chord.	Sine.	Tangent.	Co-tangent.	Cosine			
De-grees.	Radians.								
0°	0	000	0	0	∞	1	1·414	1·5708	90°
1	·0175	·017	·0175	·0175	57·2900	·9998	1·402	1·5533	89
2	·0349	·035	·0349	·0349	28·6363	·9994	1·389	1·5359	88
3	·0524	·052	·0523	·0524	19·0811	·9986	1·377	1·5184	87
4	·0698	·070	·0698	·0699	14·3007	·9976	1·364	1·5010	86
5	·0873	·087	·0872	·0875	11·4301	·9962	1·351	1·4835	85
6	·1047	·105	·1045	·1051	9·5144	·9945	1·338	1·4661	84
7	·1222	·122	·1219	·1228	8·1443	·9925	1·325	1·4486	83
8	·1396	·140	·1392	·1405	7·1154	·9903	1·312	1·4312	82
9	·1571	·157	·1564	·1584	6·3138	·9877	1·299	1·4137	81
10	·1745	·174	·1736	·1763	5·6713	·9848	1·286	1·3963	80
11	·1920	·192	·1908	·1944	5·1446	·9816	1·272	1·3788	79
12	·2094	·209	·2079	·2126	4·7046	·9781	1·259	1·3614	78
13	·2269	·226	·2250	·2309	4·3315	·9744	1·245	1·3439	77
14	·2443	·244	·2419	·2493	4·0108	·9703	1·231	1·3265	76
15	·2618	·261	·2588	·2679	3·7321	·9659	1·218	1·3090	75
16	·2793	·278	·2756	·2867	3·4874	·9613	1·204	1·2915	74
17	·2967	·296	·2924	·3057	3·2709	·9563	1·190	1·2741	73
18	·3142	·313	·3090	·3249	3·0777	·9511	1·176	1·2566	72
19	·3316	·330	·3256	·3443	2·9042	·9455	1·161	1·2392	71
20	·3491	·347	·3420	·3640	2·7475	·9397	1·147	1·2217	70
21	·3665	·364	·3584	·3839	2·6051	·9336	1·133	1·2043	69
22	·3840	·382	·3746	·4040	2·4751	·9272	1·118	1·1868	68
23	·4014	·399	·3907	·4245	2·3559	·9205	1·104	1·1694	67
24	·4189	·416	·4067	·4452	2·2460	·9135	1·089	1·1519	66
25	·4363	·433	·4226	·4663	2·1445	·9063	1·075	1·1345	65
26	·4538	·450	·4384	·4877	2·0503	·8988	1·060	1·1170	64
27	·4712	·467	·4540	·5095	1·9626	·8910	1·045	1·0996	63
28	·4887	·484	·4695	·5317	1·8807	·8829	1·030	1·0821	62
29	·5061	·501	·4848	·5543	1·8040	·8746	1·015	1·0647	61
30	·5236	·518	·5000	·5774	1·7321	·8660	1·000	1·0472	60
31	·5411	·534	·5150	·6009	1·6643	·8572	·985	1·0297	59
32	·5585	·551	·5299	·6249	1·6003	·8480	·970	1·0123	58
33	·5760	·568	·5446	·6494	1·5399	·8387	·954	·9948	57
34	·5934	·585	·5592	·6745	1·4826	·8290	·939	·9774	56
35	·6109	·601	·5736	·7002	1·4281	·8192	·923	·9599	55
36	·6283	·618	·5878	·7265	1·3764	·8090	·908	·9425	54
37	·6458	·635	·6018	·7536	1·3270	·7986	·892	·9250	53
38	·6632	·651	·6157	·7813	1·2799	·7880	·877	·9076	52
39	·6807	·668	·6293	·8098	1·2349	·7771	·861	·8901	51
40	·6981	·684	·6428	·8391	1·1918	·7660	·845	·8727	50
41	·7156	·700	·6561	·8693	1·1504	·7547	·829	·8552	49
42	·7330	·717	·6691	·9004	1·1106	·7431	·813	·8378	48
43	·7505	·733	·6820	·9325	1·0724	·7314	·797	·8203	47
44	·7679	·749	·6947	·9657	1·0355	·7193	·781	·8029	46
45°	·7854	·765	·7071	1·0000	1·0000	·7071	·765	·7854	45°
			Cosine.	Co-tangent.	Tangent.	Sine.	Chord.	Radians.	De-grees.
Angle.									

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88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0 1 1 2	2	3 3 4 4
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91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0 1 1 2	2	3 3 4 4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0 1 1 2	2	3 3 4 4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0 1 1 2	2	3 3 4 4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0 1 1 2	2	3 3 4 4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0 1 1 2	2	3 3 4 4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0 1 1 2	2	3 3 4 4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0 1 1 2	2	3 3 4 4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0 1 1 2	2	3 3 4 4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0 1 1 2	2	3 3 3 4

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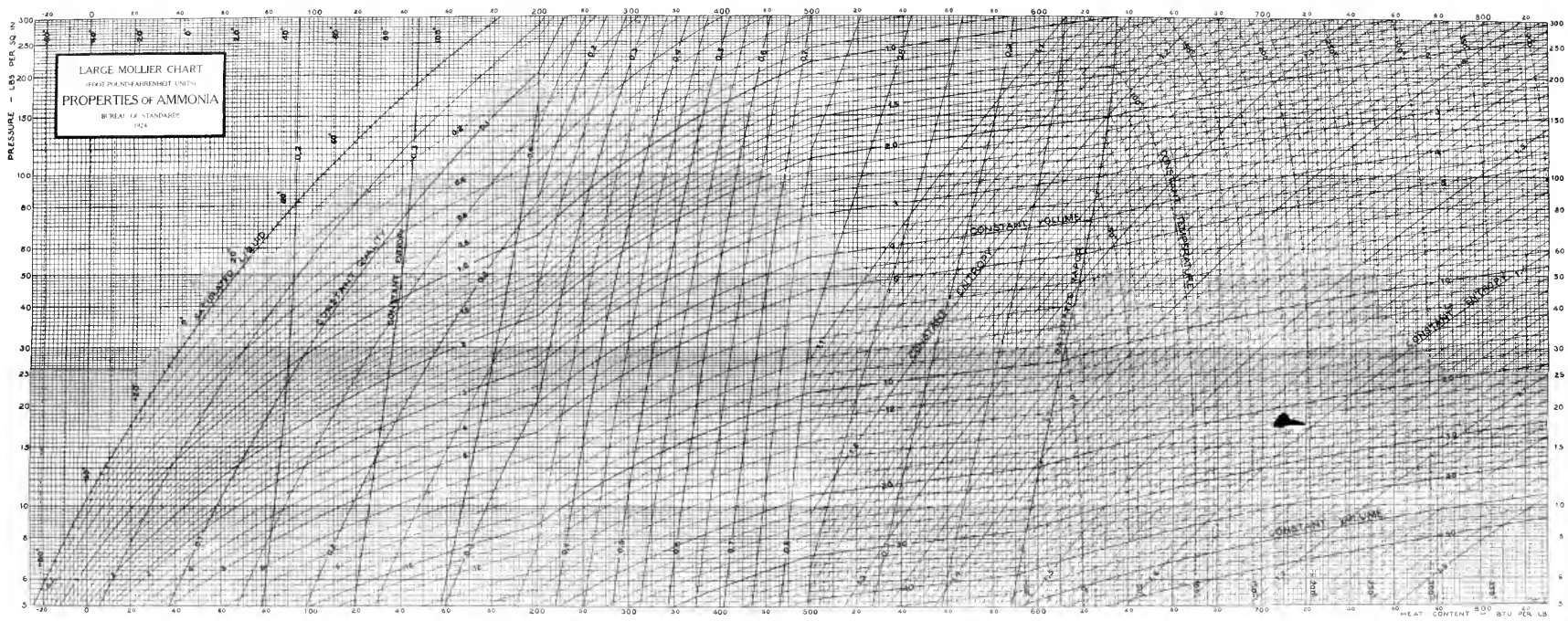
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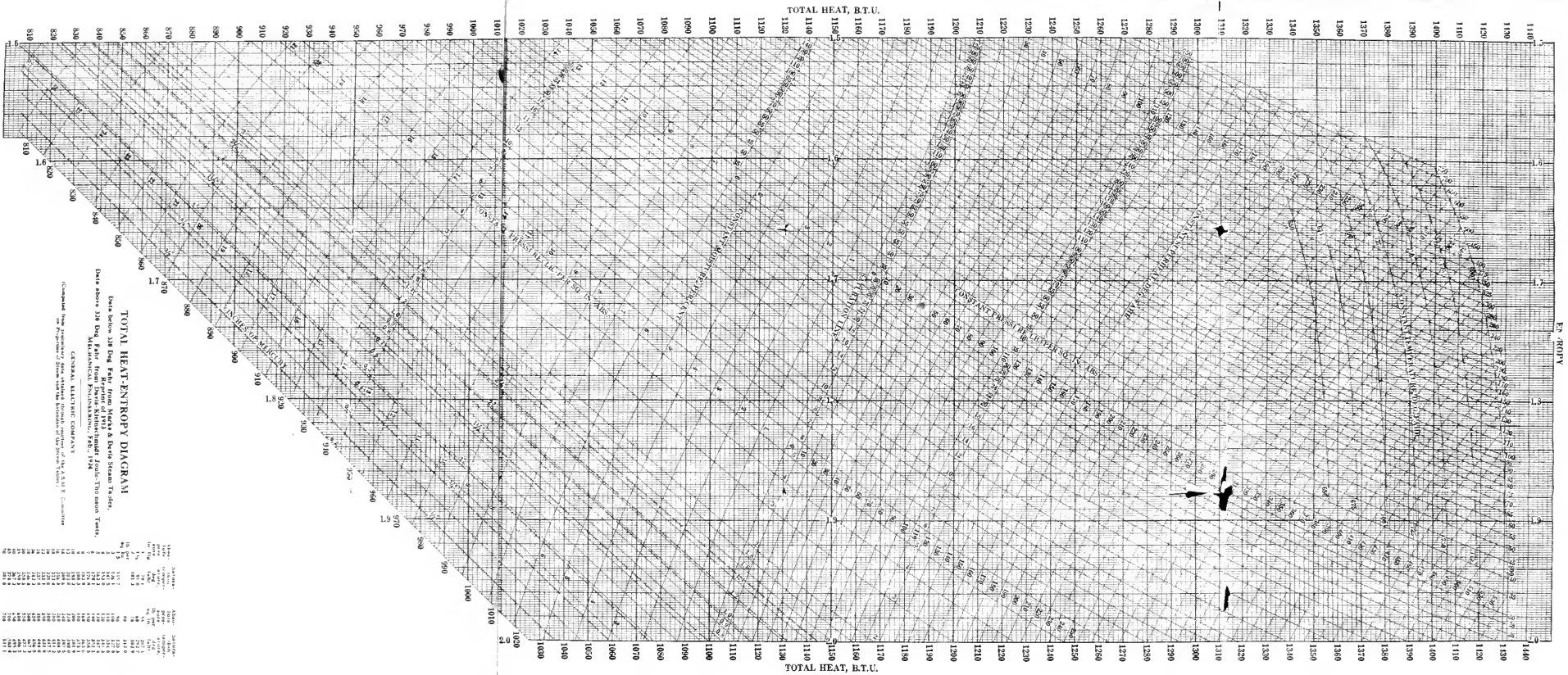
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LARGE MOLLIER CHART
(FOOT POUNDS-FAHRENHEIT UNITS)
PROPERTIES OF AMMONIA
BUREAU OF STANDARDS
1924



TOTAL HEAT-ENTROPY DIAGRAM
 Data below 318 Deg. Fahr. from Mears & Davis Steam Tables.
 Data above 318 Deg. Fahr. from Davis Steam-Engine Table-The steam Table.
 MECHANICAL ENGINEERING, 1928, 1924

GENERAL ELECTRIC COMPANY

(Compiled from preliminary data obtained from courtesy of the A.S.M.E. Committee on Properties of Steam and the Division of the American Institute of Mechanical Engineers.)

TEMPERATURE	ENTROPY	LOG. MEAN TEMPERATURE	LOG. MEAN PRESSURE	LOG. MEAN SPECIFIC VOLUME	LOG. MEAN DENSITY
100	0.171	100	1.000	1.000	1.000
200	0.315	200	1.000	1.000	1.000
300	0.459	300	1.000	1.000	1.000
400	0.603	400	1.000	1.000	1.000
500	0.747	500	1.000	1.000	1.000
600	0.891	600	1.000	1.000	1.000
700	1.035	700	1.000	1.000	1.000
800	1.179	800	1.000	1.000	1.000
900	1.323	900	1.000	1.000	1.000
1000	1.467	1000	1.000	1.000	1.000
1100	1.611	1100	1.000	1.000	1.000
1200	1.755	1200	1.000	1.000	1.000
1300	1.899	1300	1.000	1.000	1.000
1400	2.043	1400	1.000	1.000	1.000
1500	2.187	1500	1.000	1.000	1.000
1600	2.331	1600	1.000	1.000	1.000
1700	2.475	1700	1.000	1.000	1.000
1800	2.619	1800	1.000	1.000	1.000
1900	2.763	1900	1.000	1.000	1.000
2000	2.907	2000	1.000	1.000	1.000
2100	3.051	2100	1.000	1.000	1.000
2200	3.195	2200	1.000	1.000	1.000
2300	3.339	2300	1.000	1.000	1.000
2400	3.483	2400	1.000	1.000	1.000
2500	3.627	2500	1.000	1.000	1.000
2600	3.771	2600	1.000	1.000	1.000
2700	3.915	2700	1.000	1.000	1.000
2800	4.059	2800	1.000	1.000	1.000
2900	4.203	2900	1.000	1.000	1.000
3000	4.347	3000	1.000	1.000	1.000
3100	4.491	3100	1.000	1.000	1.000
3200	4.635	3200	1.000	1.000	1.000
3300	4.779	3300	1.000	1.000	1.000
3400	4.923	3400	1.000	1.000	1.000
3500	5.067	3500	1.000	1.000	1.000
3600	5.211	3600	1.000	1.000	1.000
3700	5.355	3700	1.000	1.000	1.000
3800	5.499	3800	1.000	1.000	1.000
3900	5.643	3900	1.000	1.000	1.000
4000	5.787	4000	1.000	1.000	1.000
4100	5.931	4100	1.000	1.000	1.000
4200	6.075	4200	1.000	1.000	1.000
4300	6.219	4300	1.000	1.000	1.000
4400	6.363	4400	1.000	1.000	1.000
4500	6.507	4500	1.000	1.000	1.000
4600	6.651	4600	1.000	1.000	1.000
4700	6.795	4700	1.000	1.000	1.000
4800	6.939	4800	1.000	1.000	1.000
4900	7.083	4900	1.000	1.000	1.000
5000	7.227	5000	1.000	1.000	1.000
5100	7.371	5100	1.000	1.000	1.000
5200	7.515	5200	1.000	1.000	1.000
5300	7.659	5300	1.000	1.000	1.000
5400	7.803	5400	1.000	1.000	1.000
5500	7.947	5500	1.000	1.000	1.000
5600	8.091	5600	1.000	1.000	1.000
5700	8.235	5700	1.000	1.000	1.000
5800	8.379	5800	1.000	1.000	1.000
5900	8.523	5900	1.000	1.000	1.000
6000	8.667	6000	1.000	1.000	1.000
6100	8.811	6100	1.000	1.000	1.000
6200	8.955	6200	1.000	1.000	1.000
6300	9.099	6300	1.000	1.000	1.000
6400	9.243	6400	1.000	1.000	1.000
6500	9.387	6500	1.000	1.000	1.000
6600	9.531	6600	1.000	1.000	1.000
6700	9.675	6700	1.000	1.000	1.000
6800	9.819	6800	1.000	1.000	1.000
6900	9.963	6900	1.000	1.000	1.000
7000	10.107	7000	1.000	1.000	1.000
7100	10.251	7100	1.000	1.000	1.000
7200	10.395	7200	1.000	1.000	1.000
7300	10.539	7300	1.000	1.000	1.000
7400	10.683	7400	1.000	1.000	1.000
7500	10.827	7500	1.000	1.000	1.000
7600	10.971	7600	1.000	1.000	1.000
7700	11.115	7700	1.000	1.000	1.000
7800	11.259	7800	1.000	1.000	1.000
7900	11.403	7900	1.000	1.000	1.000
8000	11.547	8000	1.000	1.000	1.000
8100	11.691	8100	1.000	1.000	1.000
8200	11.835	8200	1.000	1.000	1.000
8300	11.979	8300	1.000	1.000	1.000
8400	12.123	8400	1.000	1.000	1.000
8500	12.267	8500	1.000	1.000	1.000
8600	12.411	8600	1.000	1.000	1.000
8700	12.555	8700	1.000	1.000	1.000
8800	12.699	8800	1.000	1.000	1.000
8900	12.843	8900	1.000	1.000	1.000
9000	12.987	9000	1.000	1.000	1.000
9100	13.131	9100	1.000	1.000	1.000
9200	13.275	9200	1.000	1.000	1.000
9300	13.419	9300	1.000	1.000	1.000
9400	13.563	9400	1.000	1.000	1.000
9500	13.707	9500	1.000	1.000	1.000
9600	13.851	9600	1.000	1.000	1.000
9700	13.995	9700	1.000	1.000	1.000
9800	14.139	9800	1.000	1.000	1.000
9900	14.283	9900	1.000	1.000	1.000
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